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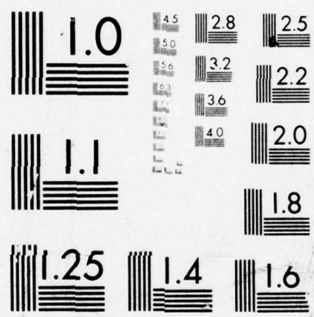

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Part 1 of 2

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FOREIGN TECHNOLOGY DIVISION

AIR FORCE SYSTEMS COMMAND

HYDRAULIC DRIVE AND
HYDROPNEUMOAUTOMATION

by

T. M. Bashta

FOREIGN TECHNOLOGY DIVISION

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HYDRAULIC DRIVE AND HYDROPNEUMOAUTOMATION

By: T. M. Bashta

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U. S. BOARD ON GEOGRAPHIC NAMES TRANSLITERATION SYSTEM

Block	Italic	Transliteration	Block	Italic	Transliteration
А а	*А а*	A, a	Р р	*Р р*	R, r
Б б	*Б б*	B, b	С с	*С с*	S, s
В в	*В в*	V, v	Т т	*Т т*	T, t
Г г	*Г г*	G, g	У у	*У у*	U, u
Д д	*Д д*	D, d	Ф ф	*Ф ф*	F, f
Е е	*Е е*	Ye, ye; E, e*	Х х	*Х х*	Kh, kh
Ж ж	*Ж ж*	Zh, zh	Ц ц	*Ц ц*	Ts, ts
З з	*З з*	Z, z	Ч ч	*Ч ч*	Ch, ch
И и	*И и*	I, i	Ш ш	*Ш ш*	Sh, sh
Й й	*Й й*	Y, y	Щ щ	*Щ щ*	Shch, shch
К к	*К к*	K, k	Ъ ъ	*Ъ ъ*	"
Л л	*Л л*	L, l	Ы ы	*Ы ы*	Y, y
М м	*М м*	M, m	Ь ь	*Ь ь*	'
Н н	*Н н*	N, n	Э э	*Э э*	E, e
О о	*О о*	O, o	Ю ю	*Ю ю*	Yu, yu
П п	*П п*	P, p	Я я	*Я я*	Ya, ya

*ye initially, after vowels, and after ъ, ь; e elsewhere. When written as ё in Russian, transliterate as yё or ё. The use of diacritical marks is preferred, but such marks may be omitted when expediency dictates.

GREEK ALPHABET

Alpha	Α	α	*α*	Nu	Ν	ν	
Beta	Β	β		Xi	Ξ	ξ	
Gamma	Γ	γ		Omicron	Ο	ο	
Delta	Δ	δ		Pi	Π	π	
Epsilon	Ε	ε	*ϵ*	Rho	Ρ	ρ	*ϱ*
Zeta	Ζ	ζ		Sigma	Σ	σ	*ς*
Eta	Η	η		Tau	Τ	τ	
Theta	Θ	θ	*ϑ*	Upsilon	Τ	υ	
Iota	Ι	ι		Phi	Φ	φ	ϕ
Kappa	Κ	ϰ κ	*ϰ*	Chi	Χ	χ	
Lambda	Λ	λ		Psi	Ψ	ψ	
Mu	Μ	μ		Omega	Ω	ω	

RUSSIAN AND ENGLISH TRIGONOMETRIC FUNCTIONS

Russian	English
sin	sin
cos	cos
tg	tan
ctg	cot
sec	sec
cosec	csc
sh	sinh
ch	cosh
th	tanh
cth	coth
sch	sech
csch	csch
arc sin	$\sin^{-1}$
arc cos	$\cos^{-1}$
arc tg	$\tan^{-1}$
arc ctg	$\cot^{-1}$
arc sec	$\sec^{-1}$
arc cosec	$\csc^{-1}$
arc sh	$\sinh^{-1}$
arc ch	$\cosh^{-1}$
arc th	$\tanh^{-1}$
arc cth	$\coth^{-1}$
arc sch	sech^{-1}
arc csch	csch^{-1}
rot	curl
lg	log

GRAPHICS DISCLAIMER

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HYDRAULIC DRIVE AND HYDROPNEUMOAUTOMATION

T. M. Basht.

Pages 1-320.
FTD-ID(RS)T-0109-77

Page 2.

Basht T. M. Hydraulic drive and hydropneumoautomation. M, "Machine-building", 1972, 320 s.

In textbook is presented the wide complex of questions concerning equipment/device, the calculation, construction and the production of volumetric hydropneumatic drives and hydropneumatic systems, and also concerning the application/use of these equipment/devices for mechanization and the automation of production processes.

In the sections, dedicated to the cell/elements of hydraulic systems, is minutely examined the equipment/device and the calculation of actuating mechanisms, and also by varied distributive, that controls, protective and other hydro- and the pneumatic equipment with the aid of which are provided the automatic control of these actuating mechanisms and the assigned law of their motion. Is

minutely examined the choke and volumetric automatic velocity control of these mechanisms and the provision for their assigned interaction. Are examined equipment/devices for the decontamination of liquid and sealing/pressurization of connection/compounds.

The properties of working fluid and gases, the laws of applied hydraulics and gas hydrodynamics are presented taking into account the specific character of hydraulic and pressure-operated devices.

The properties of working fluid and gases, the laws of applied hydraulics and gas hydrodynamics are presented taking into account the specific character of hydraulic and pressure-operated devices.

From the cell/elements of automatic and semiautomatic control (control) are comprised typical complex hydropneumatic system, are given the principles of the calculation of these systems.

Is described the device of pneumatic actuators, and also hydro- and pneumatic boosters the slave/servo type, widely utilized in the systems of azimuth guidance of cargo vehicles, in the copying machine

tools and the other machines, which require the provision for the assigned law of the motion of exit (performing) component/link. Table 2, Illustration 260, the copy of lit. 24 titles.

Page 3.

Introduction.

The present course precede courses "hydraulics", "gas hydrodynamics" and "displacment pumps and hydraulic engines". After it follow the courses the "theory of automatic control and dynamics of hydropneumatic systems", and also "technology of the production of hydraulic drive". Taking into account this sequence of research on the erumerated disciplines, is constructed the set forth in the present textbook material of course. In accordance with this, all questions, which relate to thematics of the named courses (common/general/total questions hydro- and gas dynamics, the dynamics of hydroaggregates and hydraulic systems, constructions of pumps and hydraulic engines, production of hydroaggregates, etc.), are examined in the present course only in that volume, in which this necessary for the understanding of the set forth material of course.

Terms and common/general/total determinations. By volumetric hydropneumatic drives it is understood in the general case of the hydropneumatic system, interded for bringing into the motion of mechanisms and machines, which is volumetric hydropneumatic engine.

Common also the definition, according to which by volumetric hydraulic drive is understood the hydraulic system (system of hydraulic machines and hydroaggregates), employed for a transmission by means of the liquid of energy to distance and its conversion into mechanical output energy of system (into the kinetic energy of hydralic engine) and simultaneously fulfilling the functions of control and reversing the speed of exit component/link.

The volumetric hydraulic drive, which consists of the devices, structurally designed in one common/general/total block, is called volumetric hydraulic transmissions (by hydraulic transmission). Concept "hydraulic drive" usually is identified with concept "hydraulic system" hearth by which is understood the totality of the devices, which transmit energy by using a liquid on by pressure.

Hydraulic system can have both one and several hydraulic engines and pumps.

Any hydraulic drive consists of the source of the fluid flow rate, as which in the majority of cases serves the pump, the hydraulic engine of reciprocating or rotary motion, units of control, liquid main lines (hydraulic lines or drainage systems) and of other hydraulic equipment.

By hydrobond is understood the totality of the connected with each other devices, having direct contact with working fluid and intended for the execution of the determined function in volumetric hydraulic drive.

Hydroapparatus are called the devices, intended for a change in the flow parameter of working fluid or for their maintaining at the determined fixed level. Under flow parameters in this case, understand the pressure, the expenditure/consumption and direction of

rction.

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By hydraulic lines or by drainage system is implied the device, intended for the passage of working fluid in the process of the work of volumetric hydraulic drive.

They distinguish:

pressure hydraulic line - part of the basic hydraulic line over which the working fluid moves from distributor to hydraulic engine and vice versa;

performing hydraulic line - part of the basic hydraulic line over which working fluid it moves from distributor to hydraulic engine and vice versa;

drainage gidroliniyu - part of the basic hydraulic line over which working fluid it moves tank from distributor or directly from hydraulic engine.

Pump is called the machine, which converts the mechanical energy, applied to its shaft, into the energy of liquid, and hydraulic engine she is called the machine, which converts the energy of liquid into mechanical energy on its shaft.

volumetric hydraulic engine with the rotary motion of driven/known component/link to angle of <360° is called a hydrorotor, or a moment hydraulic cylinder or hydroquadrant. Volumetric hydraulic drive with hydrorotor is called hydraulic drive of rotary motion.

Is applied also collecting term the volumetric hydraulic machine hearth by which is understood the totality of pump and hydraulic engine. Volumetric hydraulic machine, this device, intended for the conversion of the mechanical energy of the inlet into the energy of output/yield in the process of the filling of working chambers with working fluid and its displacement from these chambers, whereupon by working chamber is understood the limited space of volumetric

hydraulic machine, which periodically changes its volume and which is alternately communicated with the receiving or giving up cavity of hydraulic machine.

Volumetric hydraulic engine with the rotary motion of driven/known (exit) component/link is called hydraulic motor, and hydraulic engine with straight reciprocating motion - hydraulic cylinder.

In accordance with this, volumetric hydraulic drive in which driven/known component/link accomplishes rotary motion, is called hydraulic drive of rotary motion, and hydraulic drive in which exit component/link accomplishes rectilinear motion, by hydraulic drive of rectilinear motion.

Depending on whether does come the working medium from volumetric hydraulic engine into tank or in the suction line of pump, are distinguished hydraulic drive with open circulation (working medium enters tank) and enclosed circulation (working medium enters into the suction line of pump).

Advantages and the fields of application of hydraulic drive. The practice of the application/use of hydraulic drive in industry, and in particular in machine-building, demonstrated their progressive role in development of technology. Because of by such the important for the majority of the cases of application/use advantages of hydraulic drive as a small mass and volume, per unit the unit of transmitted power, high efficiency, the reliability of action, and also the simplicity of the automation of control the hydraulic drive had extensive application in the different branches of machine-building.

In addition to the indicated advantages the hydraulic engines of rotary action (hydraulic motors) differ in terms of the large ratio of the torsional moment on output shaft to the moment of inertia of rotor, which determines the dynamic properties of engine.

Because of the indicated favorable ratio of the torque of hydraulic motor to the moment of inertia of its moving elements, can be obtained the negligibly short time: its reverse (0.03-0.05 s), the achievements of the maximum frequencies of the rotation/revolution

(the high speed operation cf the drive) and of delay during final adjustment by the hydraulic motor of control signals.

Page 5. ln3 in view of this hydraulic drive provides the high frequency of reversings (for rotary type hydraulic motor it can be led to 500 and more reversings per minute). The number of reversings of hydraulic drive of the rectilinear motion with relatively small by mass and course reaches 1000 per minute.

In terms of high speed operation differ also pumps. Thus, for instance, time during which the feed of some pumps, in particular aviation, can be changed from the zero to the maximum values, does not exceed 0.04 s, but time of a reduction in the feed from the maximum value to the zero - 0.02 s.

The advantage of hydraulic systems is also the possibility of the infinitely variable control of the exit velocity over a wide range. The gear ratio of hydraulic drive of the rotary action hearth by which is understood the ratio of the minimum frequency of the *rotation/revolution* of the shaft of hydraulic motor to maximum, comprises in many instances of 1000. The lower frequency limit of the

rotation/revolution of the majority of the existing hydraulic motors is led to 5-10 r/min.

At the same time hydraulic drive are simple in production, operation and differ in terms of reliability - the service life of many types of pumps and hydraulic motors is led to 20 000 h and above.

Operating principle of volumetric hydraulic drive. Specific energy of an ideal liquid is determined by equation

$$e = \frac{E}{m} = zg + \frac{p}{\rho} + \frac{u^2}{2},$$

where E - total energy of liquid by density π; m is the mass of liquid, which flows at a rate of u; zg - the specific energy of position; g - free-fall acceleration; p/ρ - the specific energy of pressure; $u^2/2$ - the specific kinetic energy of liquid.

An energy transfer by liquid can be realized, by changing with any of the terms of the written above equation. In connection with volumetric hydraulic drive in question from the indicated three forms of the mechanical energy of liquid the basic form is pressure energy, which easily can be converted into mechanical work with the aid of hydraulic engines.

For auxiliary, mainly command, circuits is utilized also kinetic energy.

The kinetic energy of liquid is utilized in the hydrodynamic transmissions which are examined in analogous/similar training course. The energy of position in volumetric hydraulic drive usually they disregard, since altitude differences z between the separate cell/elements of hydraulic system are small and the energy of position is incommensurable small in comparison with the acting in it pressure energy of liquid. This energy of position is considered only during calculations and studies of the suction pump performance.

The operating principle of volumetric hydraulic drive is based on the high bulk rigidity modulus (negligible compressibility) of

liquid and on noted French scholarly B. Pascal's law which says, that any change of the pressure in any point of the quiescent true liquid, which does not disrupt its equilibrium, is transferred to other points without change.

From the given in Fig. 1A diagram, which illustrates this law, it follows that if to that which hermetically seal the filled by liquid enclosed cylindrical container to piston with an area of f we add force of P, then this force will be balanced (friction of piston we disregard) by the force of pressure of liquid p on this piston, which will act at any point of liquid, including the surface of container (hydrostatic pressure, caused by gravity strength of liquid, disregard).

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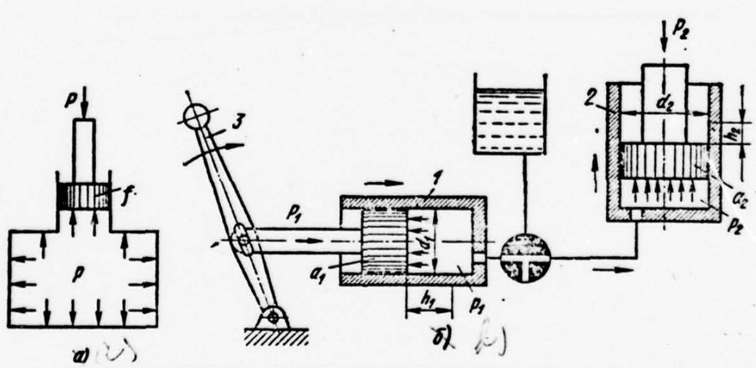

Fig. 1. Diagrams, which illustrate the operating principle of volumetric hydraulic drive.

Position will be preserved, if as container we take two connected with conduit/manifolds hermetically sealed by pistons a_1 and a_2 (Fig. 1b) cylinder 1 and 2, the first of which is that which drive (by pump) and the second - by driven/known (by hydraulic engine) component/link. In moving with the aid of the crank of 3 pistons a_1 cylinder 1 to starboard the liquid is displaced into cylinder 2, moving its piston a_2 upward, whereupon pressure

$$p_1 = \frac{P_1}{f_1},$$

developed in cylinder 1 with force P_1, applied to piston a_1, it acts also to piston a_2 cylinder 2 (losses of pressure in conduit/manifold we disregard, i.e., we consider that $p_2 = p_1$).

Assuming that cylinders 1 and 2 are hermetically sealed, but incompressible fluid, the displacement/movement of pistons a_1 and a_2 let us describe by the equation of the equality of the displaced by them volumes (by equation of flow continuity)

$$h_1 f_1 = h_2 f_2,$$

where h_1, h_2, f_1 and f_2 - respectively displacement/movements and piston clearance a_1 and a_2.

On the basis of the last/latter equality

$$\frac{h_2}{h_1} = \frac{f_1}{f_2} = \frac{d_1^2}{d_2^2};$$
$$h_2 = h_1 \frac{f_1}{f_2} = h_1 \frac{d_1^2}{d_2^2},$$

where d_1 and D_2 are diameters of pistons a_1 and a_2.

ab Disregarding hydraulic friction (assuming that pressure $p_1 = p_2 = p$) and friction of pistons a_1 and a_2 during their motion, it is possible to write expressions for forces P_1 and P_2, which act on pistons a_1 and a_2:

$$P_1 = pf_1 \text{ and } P_2 = pf_2.$$

In accordance with this

$$\frac{P_2}{P_1}=\frac{pf_2}{pf_1}=\frac{d_2^2}{d_1^2};\ P_2=P_1\frac{f_2}{f_1},$$

where p - the pressure of liquid in cylinders; P_1 and P_2 are forces of pressure of liquid respectively on pistons a_1 and a_2.

From this equality it follows that force P_2 is greater than force P_1 in f_2/f_1 once.

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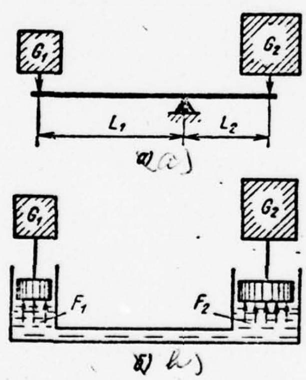


Fig. 2. Diagrams, which illustrate the analogy between mechanical and hydraulic transmissions.

Equilibrium of forces, which act in the examined diagram, can be compared with the equilibrium of the yoke/arm, loaded by loads with a weight of G_1 and G_2, applied on its ends (Fig. 2a).

It is not difficult to see that the reaches L_1 and L_2 yoke/arm and the value of loads with a weight of G_1 and G_2 are connected by relationship

$$\frac{G_1}{G_2} = \frac{L_2}{L_1}.$$

Respectively for the compared hydraulic circuit (Fig. 2b), which is of the connected with corduit/manifolds two cylinders with an area of F_1 and F_2 whose porshin are loaded by loads with a weight of G_1 and G_2,

$$\frac{G_1}{G_2} = \frac{F_1}{F_2}.$$

the product of force p_1, which acts on piston a_1 (see Fig. 1b), to the velocity of its motion

$$v_1 = \frac{h_1}{t},$$

where t - time of the displacement/movement of piston up to distance h_1, will give the expression of power

$$N = P_1 v_1.$$

After substituting into previous expression $P_1 = f_1 p$, we will obtain

$$N = pf_1 v_1.$$

Taking into account that product $f_1 v_1$ expresses the volume, described by piston per unit time, or the calculated feed Q of liquid by piston, we will obtain

$$N = pQ. \qquad (1)$$

In such a case, when pressure is expressed in kgf/cm^2 feed in cm^3/s, power will be expressed in kgf•cm/s.

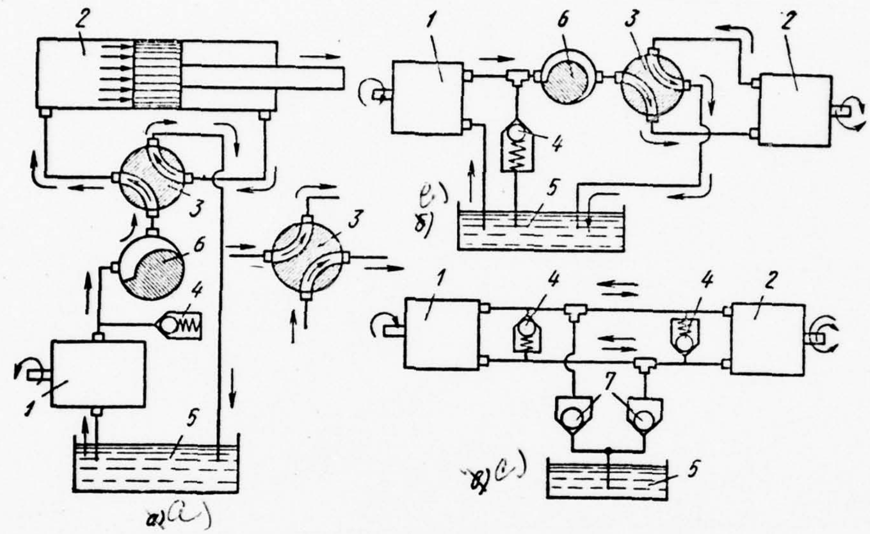
For obtaining power in h.p., they use expression

$$N = \frac{pQ}{7500} \text{ h.p.,}$$

where Q they use the fluid flow rate in cm^3/s; p is pressure of liquid in kgf/cm^2.

Design concept differs from the simplified schematic diagram examined above (see Fig. 1b) in terms of the fact that it includes continuous pump, and also a series of the supplementary apparatuses which make it possible to control fluid flow, which enters from pump the hydraulic engine, and to protect system from overloads.

Fig. 3. Schematic diagrams of hydraulic drive of rectilinear (a) and rotary (b and c) motions.



In accordance with this, distinguish in any hydraulic drive three groups of the cell/elements: pumps (sources of hydraulic power), hydraulic engines (receivers of hydraulic power or actuating mechanisms), the distributive and controlling hydroapparatus.

Figure 3a depicts the diagram of the simplest volumetric hydraulic drive for straight reciprocating motion. Drive consists of those connected with the conduit/manifolds of pump 1 with reservoir (tank) 5 and hydraulic engine (actuating cylinder) 2, safety valve 4, which restricts a pressure increase of liquid higher than rating value, and the distributor (tap/crane) 3 with the aid of which changes the direction of fluid flow from pump to the working cavities of hydraulic engine, i.e., it is realized a change in the direction of its motion. In the position of distributor (tap/crane) 3, presented in Fig. 3a, the liquid from pump 1 enters the left cavity of cylinder 2, moving its piston to starboard.

Liquid, displaced in this case by piston from the right (nonoperative) cavity of cylinder 2, is driven out along discharge leads and the channels of distributor 3 into reservoir 5. During the rotation of distributor 3 through angle of 90°, liquid from pump 1 enters the right cavity of cylinder 2 and is abstract/removed into

tank 5 of its left cavity; piston in this case is moved to left side. During a pressure increase of liquid over rating value, will be discovered safety valve 4 and liquid under pressure it will be recasted through it into tank. Figure 3b and c depicts the schematic diagrams of hydraulic transmission with hydraulic engine (hydraulic motor) of 2 rotary motions.

The reverse of hydraulic engine in diagram in Fig. 3b is realized with the aid of distributor 3, and in diagram in Fig. 3c - by change by the pump of 1 direction of fluid flow. System in the latter case must be equipped by check (locking) valves 7, which detach during changes in the direction of fluid flow main pressure line from tank 5 and simultaneously provide the makeup of the suction cavity of pump if in the latter is formed as a result of the hydraulic slipes or other reasons vacuum. Diagram is also equipped by safety valve 4 and by the tank of 5 supply of working fluid.

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It is obvious that under the previously accepted condition of the complete airtightness of hydroaggregates and practical

incompressibility of liquid the exit component/link of engine is moved (or rotates) at the definite velocity, which ensures the passage through its working chambers of the liquid, applied by pump, i.e., must be observed the condition of the $Q_н = Q_д$, where the $Q_н$ and $Q_д$ are theoretical feeds (volumes, described by working cell/elements per unit time) of pump and engine. As a result under the previously accepted condition, we will obtain the rigid kinematic constraint between the pump and the hydraulic engine.

The velocity control of hydraulic engine (piston stroke of actuating cylinder or shaft of hydraulic motor) in transmissions by power more than 5-10 h.p. usually is realized by a feed control of pump 1 by changing his working volume, also, in the transmissions of less powers - by means of throttle/choke 6, with the aid of which is created the friction at the inlet of liquid into the hydraulic engine, as a result of which part of the liquid is recasted through safety valve 4 into tank 5. During the complete overlap of conduit/manifold by throttle/choke 6 entire liquid is recasted through valve 4 into tank, as a result the velocity of hydraulic engine 2 will be equal to zero.

From that which was given it follows that the throttle control

is connected with power loss and the heating of liquid, since the energy of the volume of the liquid of the aaaaa, jettisoned per unit time in valve 4 into tank under pressure is converted into heat.

The lost in this case power

$$N_{nom} = p_{KA} Q_{KA},$$

where the p_{KA} - the pressure to which is adjusted the safety valve 4.

The examined in Fig. 3 diagrams they are related to the number simplest, in which the control of fluid flow is reduced only to a change in its direction of flow without any effect on the law of the piston stroke of hydraulic engine. In the case when this effect is linked with the cycle of the work of machine or with control according to program, hydraulic drive becomes part of the system of automatic either semiautomatic control, the cell/elements of hydraulic drive are called the cell/elements of automatic hydraulic equipment, and hydraulic system - by automatic or semiautomatic.

Pressure of liquid in hydraulic systems. From expression (1) it

follows that during a pressure increase of liquid the power of hydraulic drive under otherwise equal conditions proportionally raised, and consequently, descend its specific mass and dimensions. In view of this in practice, occurs a continuous pressure increase. At present in hydraulic systems, as a rule, pressures, are equal to 200-250 and thinner than 350-700 kgf/cm².

Water with 20°C is converted into solid at pressure 8400 kgf/cm².

In the hydraulic systems of machines, are common the pumps by power to 75 kW (100 h.p.); however, in certain cases, and in particular in heavy machine building, they are applied pumps with driving/homing power above 3000 kW (at pressures 220 kgf/cm²).

Unity of the physical quantities. Calculation formulas in textbook, as a rule, are given in such form, that it is possible to utilize them under conditions of applying concrete/specific/actual ones of international system (SI) and of mks system, of mk forces systems and of CGS, in connection with which in the interpretation of the designations of values are not brought one of the physical

quantities.

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In those individual cases when in formulas are applied extrasystemic one as horsepower, calorie, etc., or one from different systems, in the interpretation of the designations of values, are given one, used in these calculation formulas.

In appendix are given the values of units different from ones SI and used in textbook, in ones of SI, multiple and lobate from them.

In accordance with the fact, in which ones will be examined by the reader the physical quantities (SI, µs, mk (force) s or CGS) during the analysis of formulas or the solution of the problems, given in textbook, one should apply the dimensionality, the designations and the designations, provided by one Gost or the other, namely: by project GOST [~~99sp04~~ ГОСТ - All-union State Standard] in SI publ. 1970, GOST of 9867-61 or GOST 7664-61.

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Chapter I.

PRINCIPLES OF APPLIED HYDRAULICS.

WORKING FLUIDS AND THEIR PROPERTIES.

Working medium (medium) in hydraulic systems are in essence the different types of the mineral liquids, which are the distillate oils, thickened by solid hydrocarbons (paraffin/kerosene, ceresin-80, etc.), and less frequently liquids on the basis of organic and organosilicon compounds. Especially widely are applied the mixtures of mineral oils, which consist of low-viscosity oil-products with high-viscosity components (thickeners).

The working fluid is the cell/element of hydraulic mechanism and

simultaneously lubricant and anticorrosive, therefore when selecting working fluid for hydraulic systems, one should consider its physical and chemical properties. The basic evaluation criteria of the quality of working fluid are the density, the viscosity-temperature properties, chemical and physical stability, aggressiveness with respect to rubber packing elements and lubricating power. Sometimes are presented the requirements for refractoriness and suitability of work in wide temperature range.

Density of liquids.

The density of liquid is the physical quantity, which represents the ratio of the mass of liquid to its volume.

During the even distribution of mass density

$$\rho=\frac{m}{V},$$

where m is a mass of the volume of V liquid in question.

Is distinguished also specific volume (volume, occupied by the unit of mass) v, which is the value, reciprocal densities ρ

$$v=\frac{1}{\rho}.$$

Specific gravity/weight can be expressed by density and free-fall acceleration g:

$$\gamma = \rho g.$$

The density of mineral oils varies for their different brands within limits

$$\rho = 830 \div 940 \text{ kg/m}^3.$$

For practical calculations it is possible to accept ρ = 900 kg/m³.

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Density has great significance during calculations of flow conditions of the liquid through the local friction, loss of pressure in which is caused in essence by the acceleration of liquid, and consequently, the pressure differential Δp, as this follows from known relationship

$$u=\sqrt{\frac{2\Delta p}{\rho}},$$

it depends on the density of liquid

$$\Delta p = \frac{\rho}{2} u^2.$$

On the density of liquid, depend the impact pressure with hydraulic impact, and also line resistance in transient processes. For example, for creation in the pipeline of certain in practice real acceleration of liquid with ρ = 13 600 kg/m³ the required pressure can exceed 17 times the pressure, necessary in the case of applying a mineral oil with ρ = 800 kg/m³. During the application/use of a liquid with ρ = 13 600 kg/m³ the force of inertia during its motion in conduit/manifolds will be the so large that for the creation of the required acceleration will be expend/consumed the considerable part of the pressure; with respect will decelerate the operating speed of system and the reaction of the latter to control signals.

The density of liquid depends on temperature, the form what with

change latter changes also the specific volume of liquid. The indicated dependence is characterized by temperature coefficient α (in 1/deg) volume expansion of liquid, that are the physical quantity, which expresses a relative change in the volume of liquid during a change in the temperature on 1°C:

$$\alpha = \frac{\Delta V / V_0}{\Delta t},$$

where $\Delta V/V_0$ and the relative change in the initial volume V_0 in QUESTION liquid; $\Delta t = t - t_0$ - the change of temperature; here t_0 and t are the initial and final temperatures of liquid; $\Delta V = V - V_0$ - volume change during an increase in the temperature with t_0 to t; here V_0 and V are a volume of liquid respectively at temperatures t_0 and t.

In accordance with this change ΔV volume and the new volume V at temperature of t

$$\Delta V = \alpha \Delta t V_0; \qquad (2)$$

$$V = V_0 + \Delta V = V_0 (1 + \alpha \Delta t). \qquad (3)$$

The density of liquid at rated temperature of $t = t_0 + \Delta t$

$$\rho = \frac{\rho_0}{1 + \alpha \Delta t},$$

where ρ is density of liquid at temperature of t.

The average value of temperature volumetric expansion coefficient for the widespread in hydraulic systems oil of AMG-10 can be taken as in the range of pressures 0-200 kgf/cm² $8 \cdot 10^{-4}$ to 1/deg, or, otherwise, the temperature expansion of this oil comprises

approximately 0.08o/o during heating cn 1°C. For the mineral oils of the more high viscosities, distributed in the hydraulic systems of cther machines, this coefficient is equal approximately 7•10⁻⁴ to 1/deg.

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For this case of expression (2) and (3) they take the form

$$\Delta V = 7\cdot 10^{-4}\Delta t V_0;$$
$$V = V_0(1 + 7\cdot 10^{-4}\Delta t).$$

The maximum value of temperature vclumetric expansion coefficient they have synthetic liquids. Thus, for instance, average temperature volumetric exparsicn coefficient cf liquid on the basis of alkyl polysiloxates during a change in the temperature from 0 to 200°C is equal to 9.52•10⁻⁴ to 1/deg.

Since the density of true liquids changes with a change in the temperature in the widespread temperature range insignificantly, during the hydraulic designs in many instances sufficient to take the constant values of these parameters. However, are possible conditions in which this assumption it can give as a result of the volume expansion of liquid during a change in its temperature to the serious disruptions of the work of hydraulic system. The latter is caused by the fact that as a result of heating liquid can occur the overfilling by it of reservoirs. In the same case when liquid is included in the rigid closed capacitance/capacity (reservoir, actuating cylinder, etc.), is probable the destruction of the latter. The possibility of similar destruction is caused by the difference in the value of the temperature volumetric expansion coefficient of liquid and metals, in consequence of which in closed volumes liquid during its heating, they can arise inadmissibly high pressures. A pressure increase Δp during heating actuating cylinder (or another tank) with the closed in it liquid during a change in the temperature from t_1 to t_2, will compose

$$\Delta p = E(\alpha_{ж} - \alpha_{м})(t_2 - t_1),$$

where E is the bulk rigidity modulus of liquid; $\alpha_{ж}$ and $\alpha_{м}$ are

temperature volumetric expansion coefficients cf liquid and metal from which is made cylinder.

The density of liquid depends alsc as a result of its compressibility on the value of pressure.

However, since for the widespread working fluids with the bulk mcdulus cf compression E = 15 000-20 000 kgf/cm² density ρ at pressures on the order of 200 kgf/cm² insignificantly differs from density ρ_0 at zero pressure (virtually $\rho = 1.01\rho_0$), during the calculation of hydraulic systems, usually they assume that the density does not depend on pressure.

Viscosity of liquids.

The viscosity cf the working fluid whose hearth is understood property to resist shearing strain or the slip cf its layers, is one cf most important for calculation and design of the volumetric hydraulic equipment of the parameters cf liquid.

The mechanism of the emergence of viscosity is caused by the fact that during flow of liquid alcng sclid wall the velocity of its layers as a result of braking flow is different, in consequence of which between layers appears the fricticnal force. This frictional force (shearing stress) is determined from the equation, which expresses the law of liquid friction of Newton [19]:

$$T=\mu F\frac{du}{dy};\quad \mu=\frac{T}{F}\cdot\frac{1}{du/dy}, \tag{4}$$

where μ it is determined factor of proportionality (dynamic viscosity cf liquid); F is an area of the surface of liquid or wall in question, which is contacted with liquid; du/dy is a velocity gradient of velocity; here y - the distance between the layers of liquid, measured perpendicularly to the directicn of the motion of liquid; u - the velocity of the motion of liquid.

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From formula (4) it follows that the dynamic viscosity is numerically equal to the frictional force, which develops on single surface with the velocity gradient of velocity, equal to unity. Furthermore, from this same formula it follows that viscosity (with the exception of anomalous liquids, to which are related suspensions, colloids, etc.) it is exhibited only during flow of liquid, whereas in the quiescent liquid shearing stresses are equal to zero.

Unity of dynamic viscosity in mks system and SI - Pascal-second (Pa•s); in - MKGSS system kilogram-force-second per square meter (kgf•s/m²); in CGS system - poise (poise), equal to dyne-second per square centimeter (dyn•s/cm²). 1 kg•s/m² = 98.0665 poises = 9.80 655 Pa•s, 1 poise = 0.1 Pa•s 0,0102 kg•s/m².

Dynamic viscosity for low-viscosity liquids usually express in centipoises (cp), whereupon 1 cp = 0.01 poise. For a demonstrative comparison it is possible to indicate that the viscosity of water with 20°C is equal approximately 1 cp.

Kinematic viscosity. In the hydraulic designs is applied the ratio of dynamic viscosity μ tc density ρ liquid, which is called kinematic viscosity and is designated the ν:

$$\nu=\frac{\mu}{\rho}.$$

Unity of kinematic viscosity m^2/s in mks systems, SI and mk (fcrce) s, cm^2/s in cgs system.

The viscosity, equal tc 1 cm^2/s, is called stoke (steel). In technclogy will win acceptance the centistckes (c. st.), whereupon

1 c. st. = 0.01 steel = 1 mm^2/s = 10^{-6} m^2/s,

1 m^2/s = 10 000 steels = 1 000 CCC c. st.

On the standard of the USSR, the viscosity of oil is given at temperature of 50°C, in accordance with which in technical specifications it is indicated (if are absent special stipulations) the kinematic viscosity, expressed in centistokes at temperature of 50°C.

Conditional units of viscosity. In practice they use also the unit of relative viscosity, measured by the viscosimeter whose work is based on the discharge of the liquid through the metering hole of the determined diameter. Specifically, in Soviet industry is applied the Engler viscometer with the aid of whom is determined time t of discharge under dead weight 200 cm^3 of the experience/tested liquid from the cylindrical container through the assigned opening/aperture (2.8 mm in diameter) at the given temperature. Time t is compared with time of the aaaa of discharge from the same container 200 cm^3 of water at temperature of 20°C. In accordance with this, the viscosity of liquid in the degrees of relative viscosity (°VU) is expressed by relation

$$\text{ВУ} = \frac{t}{t_0}.$$

Relative viscosity frequently is expressed also with Engler's seconds, which show time of the discharge of the determined volume of liquid from the indicated viscosimeter in seconds.

Engler viscometer is applicable for liquids with viscosity not less than 1.1°VU.

The conversion of ones of relative viscosity into ones kinematic is conducted for conditional viscosity °VU on the table, given in GOST 33-66, relative viscosity more 16°VU according to formula

$$\nu_t = 7{,}4 \cdot 10^{-6} {}^\circ\text{ВУ},$$

where the ν_t it are conducted kinematic viscosity in m²/s by temperature of t.

We use the following formulas to determine the dynamic viscosity in terms of dynamic viscosity:

$$\mu = \nu\rho.$$

Fig. 4. Curve/graph for the recalculation of the dynamic viscosity of liquid into conditional.

Key: (1). Dynamic viscosity. (2). Viscosity.

Fig. 5. Dependence of the viscosity of the widespread oils on temperature.

Key: (1). Kinematic viscosity. (2). Temperature.

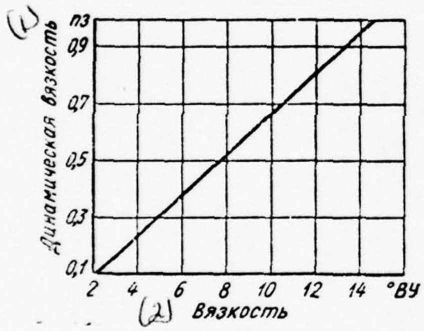

Fig. 4.

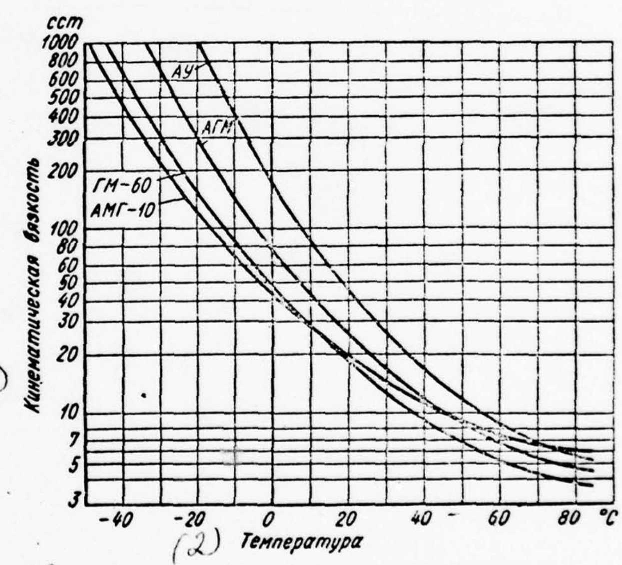

Fig. 5.

In practice for a recalculation, usually they use special nomograms or tables. Figure 4 gives curve/graph for the recalculation of dynamic viscosity into conditional.

In the USA and England, are common ones of relative viscosity - saybolt's second. The conversion of these ones into ones, accepted in Soviet industry, is conducted on special tables [7].

With the mixing of several mineral oils of different viscosity, are formed the uniform mixtures, in which are inherent the base properties of the initial oils. This property makes it possible to mix the determined amounts several types of oils for obtaining the mixture, which possesses the predetermined prevailing property, important for these target/purposes.

The viscosity of such mixture depends on the viscosity of the components and their content in mixture; however, it somewhat below average the value of the viscosity of components.

Dependence of viscosity from temperature and pressure. With an

increase in the temperature, the viscosity drop of liquids and their mixtures is reduced. Figure 5 gives the curves of the dependences of the viscosity of the used in hydraulic systems mineral oils on temperature.

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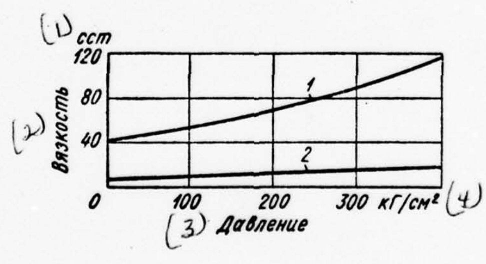
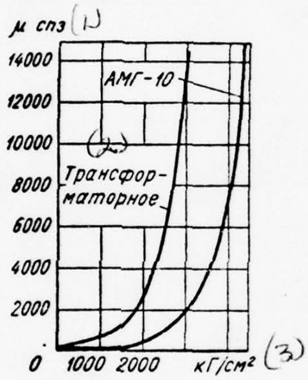


Fig. 6. Dependence of the viscosity of mineral oil on the pressure: curve 1 - 40°C; curve 2 - 80°C.

Key: (1). c. st. (2). Viscosity. (3). Pressure. (4). kgf/cm².

Fig. 7. Dependence cf the viscosity cf frostprocf mineral oils on pressure.

Key: (1). cp. (2). Transformer. (3). kgf/cm².



It is obvious, the lesser changes the visccsity with temperature change, the higher the quality and better the performance properties cf working liquid. During the applicaticn/use cf liquids with high viscosity-temperature dependence, is impeded the work of hydraulic system in winter operating conditions. The latter is caused in the basic the fact that at lcw temperatures is raised the viscosity of liquid, which makes its pumpability worse in main lines.

The viscosity cf liquids depends also cn pressure, increasing with an increase in the latter, which is substantially important for high-pressure hydraulics. Viscosity in this case change can have a considerable effect on the characteristics of hydraulic system, since even during relatively small changes in the pressure from 0 to 400 kgf/cm² the viscosity of many oils at ncrmal temperature increases approximately 3 times.

Experiment shows that at relatively small pressures (from 0 to 4CC-500 kgf/cm²) the viscosity of mineral cils changes with pressure change virtually linearly (Fig. 6). At higher pressures linear dependence is disrupted. Thus, for instance, during a pressure increase from 0 to 1500 kgf/cm² the viscosity cf mineral oils is raised 15-17 times, and by pi a pressure increase from 0 to 2000

kgf/cm² it is raised depending on the type of oil 50-1000 times. The viscosity of synthetic liquids changes under these conditions 15-25 times. At pressures on the order of 20 000-30 000 kgf/cm² mineral oils harden. In particular the oil AMG-10 hardens at pressure 30 000 kgf/cm² (at temperature of 20°C).

During practical calculations the dependence of the viscosity of the mineral oils, used in hydraulic systems, on pressure (for a range from 0 to 500 kgf/cm²) it is possible to compute according to the approximate empirical expression

$$\nu_p = \nu (1 + kp),$$

where the ν_p and ν - kinematic viscosity at pressures with respect to p and atmospheric; k is the coefficient, depending on the type of oil (it is possible to accept for the light oils with of ν_{50} <15 cSt, k = 0.002 and for the heavy oils with of ν_{50} >15 cSt, k = 0.003); p is oil pressure in kgf/cm².

Figure 7 gives the curves of the dependences of the viscosity of two widespread frostproof types of oils on pressure with temprature 20°C.

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Stability of the characteristics of oils.

For the work of hydraulic systems, it is required in order that liquids under working conditions of application/use and storage would not change their initial physical and chemical properties, i.e., retained under operating conditions physical and chemical by stability it is understood the ability of liquid to retain its physical state (properties), also, by chemical stability - stability against the "aging" hearth by which, in turn, are understood mainly the changes, which proceed in oil in the presence of oxygen of atmospheric air.

One of the reasons for the disturbance/breakdown of the physical stability of liquid is its exhaustion in continuous operation under conditions high-pressure (with high shear stresses), in particular with throttling with a large pressure differential and with lubricant under the pressure being rubbed vapor with high specific load. As a

result of this, occur the changes (destruction) in the molecular structure of liquid, which are accompanied by fall in its viscosity, and also by deterioration in its lubricating qualities.

Loss of viscosity especially strongly is exhibited of the oil mixtures of AMG-10, which contain the tougheners (thickener), which consist of long hydrocarbon chain/networks. These chain/networks with prolonged exhaustion in particular during the repeated extrusion of liquid under high pressure through small clearances, can be destroyed, occurred seemingly gradual "grinding" of thickener, as a result of which the viscosity of liquid in the course of time can decrease to inadmissibly low value.

In operation do not allow/assume reduction the viscosity more than to 20o/o of its initial value.

Not the less important characteristic of liquid is its chemical stability which depends on the chemical composition and the structure of its constituting components. As a result of the oxidation of liquids, and in particular mineral oils, occurs the precipitation them them deposits in the form of resins, and also the fall in the

viscosity, which is accompanied by the loss of lubricating properties.

The intensity of oxidation considerably depends on temperature on the contact surface of liquid with air, being raised with an increase of temperature. For example during an increase in the temperature by each 8-10°C intensity of the oxidation of mineral oil virtually is doubled.

It is obvious, in order that will occur the oxidation of oil, it must enter the contact with oxygen which occurs over bounding surface, and also over the surface of air bubbles, which is located in mechanical mixture with oil. Especially actively occurs the oxidation process of oil in the presence in it of the emulsified (undissolved) air.

Dissolution in the liquids of gases. All liquids possess the ability to dissolve gases which in the dissolved (dispersed) state do not exert a substantial influence on the operation of hydraulic system. However, if pressure at any point of the volume of liquid decreases, gases are separate/liberated from solution/opening in the

form of the bubbles which make the mechanical properties of liquid worse and reduce its chemical stability.

Experiments show that the relative volume of gas which can be dissolved in liquid before its saturation, directly proportional to face pressure of section. This volume of gas, referred to atmospheric pressure (760 mm Hg) and temperature 0°C

$$V_s = kV_{ж}\frac{p_2}{p_1},$$

where the $k = \frac{V_s}{V_{ж}} \cdot \frac{p_1}{p_2}$ - the coefficient of solubility of gas in liquid; $V_{ж}$ - the volume of liquid; p_1 and p_2 - the initial and stagnation pressure of gas, which is located in contact with the liquid.

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Fig. 8. Dependence of the relative viscosity of mineral oil on the content in it of the undissolved air.

Key: (1). Content of air bubbles.

Under coefficient of k of the solubility of gas of standard conditions, is understood the volume of the gas, which is dissolved at atmospheric pressure per unit of volume of liquid.

This coefficient depends on the properties of liquids and gases, decreasing, as a rule, with an increase in the density of liquid. For mineral oils 90 and 82 kg/m³ density the coefficient of solubility k of air with respect is 0.08 and 0.10. The coefficient of solubility of air in the oil of AMG-10 of temperature of 20°C is 0.1038 and kerosene - 0.127.

The solubility of oxygen in liquids higher, the atmospheric air, in view of which the dissolved in liquid air contains oxygen to 40-50o/o more, the atmospheric air, which intensifies the oxidation of liquid and the failure of the rubber parts of hydroaggregates.

Since the volume of the air, dissolved in liquid before its saturation, directly proportional to pressure, during a decrease (even local) in the last/latter below value at which occurred the saturation, the excess of air for a new pressure it will be isolated from it. Similar local decompression can occur as a result of a

change in velocity and direction of fluid flow in hydroaggregates and communications of hydraulic system. Gas evolution will occur until sets in the new equilibrium between liquid and gas phases.

The examined property of liquid has great practical value for the work of hydraulic system, since the presence of gas impairs, and in many instances it can completely upset the operation of hydraulic system and its aggregate/units. Specifically, when, in the liquid, gas is present, is accelerated the onset of cavitation. The gas, which isolated from liquid in the suction line of pump, can partially or even completely fill the working cavities of pump, decreasing thereby its feed and making the mode/conditions of its work worse.

Mechanical mixture of air with liquids. Air can be located in liquid in mechanical mixture with it, whereupon depending on the size/dimensions of the bubbles of the latter this mixture possesses less or larger stability, and under specific conditions, characterized in essence by the size/dimensions of bubbles (diameter of bubble is equal to 0.4-0.8 μm) and by the viscosity of liquid, the velocity of the displacement of air bubbles from liquid becomes so small, that the air bubbles can be located in mixture with liquid during many days.

The dispersal system, which consists of true liquid with gas bubbles, is called gas-liquid medium. Since gas in the form of bubbles always is present in this or another amount in the working fluids of hydraulic systems, virtually we deal not with liquid, but with gas-liquid medium. Usually in the oil of the acting hydraulic system, it is contained approximately 0.5-5c/o of air in the undissolved state.

When, in the liquid, the undissolved air is present, its viscosity increases (Fig. 8), the size/dimensions of bubbles the viscosity of mixture not affecting.

The relation of the viscosity of the liquid of μ_e with the air bubbles to the viscosity of liquid μ_0 without bubbles

$$\mu_e/\mu_0 = 1 + 0{,}015B,$$

where B is the content of air bubbles in o/c.

The presence in the liquid of disselved air considerably lowers volumetric strength of liquid and changes its cavitation properties.

Formation/education of froth. Under the known operating conditions of oil, can be formed the froth, which is the compound of air bubbles. During the formation/education of froth, occurs a fall in the lubricating qualities of oil, and also is raised the corrosion of the metallic parts of hydroaggregates and the oxidation of the oil itself, whereupon as a result of the large phase contacting area between liquid and air in froth considerably are accelerated oxidation and other chemical reactions.

The foaming and the properties of froth depend on the type of the liquid: mineral oils give stable froth, and the castor oil, which possesses the same viscosity and surface tension, easily destroys froth.

~~end section~~.

Pages 19-38.

Compressibility of liquids.

The true liquids are the elastic bodies, which are subordinated with certain assumption to the law of the compression of Hooke.

The elastic deformation (compressibility) of liquid is the

phenomenon for hydraulic drive and systems, as a rule, negative. Specifically, in view of the irreversibility of energy, expended for the compression of liquid (accumulated during the compression of liquid energy not I could be used for the completion of effective work, but is lost during expansion), is reduced the efficiency of hydraulic drive. Furthermore, the elasticity of liquid makes the operating mode of hydraulic system worse, whereupon if in slow-acting systems, and also in systems with small friction and light inertia load the effect of the elasticity of liquid usually insignificantly, then in systems with considerable inertia load and high static friction (such, for example, as systems of the drive of the tables of the machine tools) the elasticity of liquid leads to intermittent motion and the possible loss of stability.

The compressibility of liquid in the hydraulic systems of control creates in all cases in main lines and hydraulic mechanisms the effect of liquid spring.

The hardness of hydraulic mechanism it is possible to evaluate (not allowing for structural distortion) by the coefficient of relative volumetric compression β (by coefficient of compressibility), which when the compression of liquid obeys Hooke's

law itself, characterizes the relative change in the volume of liquid, per unit the unit of pressure change:

$$\beta=\frac{1}{\Delta p}\cdot\frac{\Delta V}{V_0}$$

or

$$\Delta V=\beta\,\Delta p V_0;\quad V=(V_0-\Delta V)=V_0(1-\beta\,\Delta p), \tag{5}$$

where $\Delta V/V_0$ is a relative change in the volume; $\Delta p = p_2 - p_1$ - a change in the pressure, which acts on liquid;

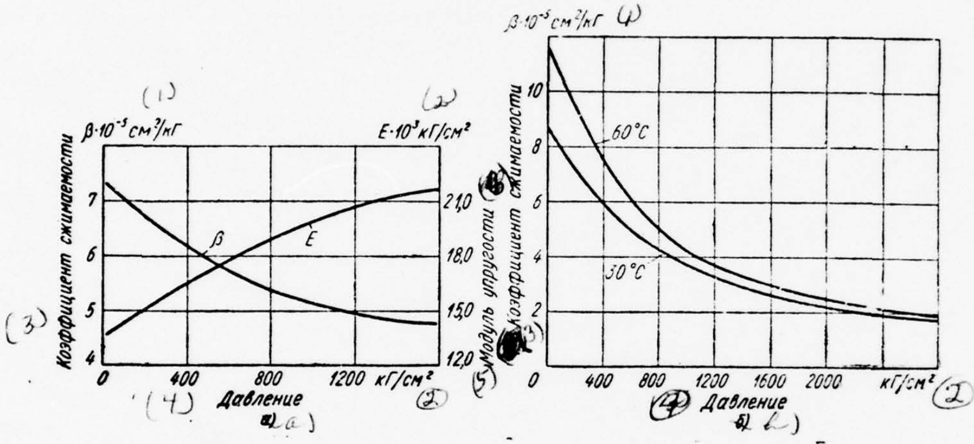
here p_2 and p_1 are the final and initial pressure; V_0 and V is the initial volume during a change in the pressure on Δp; $\Delta V = V_0 - V$ is a change in the volume of liquid with the change of pressure on Δp.

The value, reciprocal β, is called the bulk rigidity modulus of liquid during the cubic compression:

$$E=V_0\frac{\Delta p}{\Delta V}=\frac{1}{\beta}.$$

Fig. 9. Dependence of the coefficient of compressibility β and of bulk modulus E on the pressure: a) mineral oil; b) synthetic liquid.

Key: (1). cm²/kg. (2). kgf/cm². (3). Coefficient of compressibility. (4). Pressure. (5). Elastic modulus.



The bulk modulus of liquid E is accepted in technical specifications and records as the criterion for the compressibility of liquid. The value of this index depends on the type of liquid, and also on the acting pressures and temperature.

Temperature effect. With a pressure increase, the coefficient of compressibility β liquids decreases (modulus of elasticity E is raised), whereupon a decrease in it with increase pressure unevenly. For the majority of mineral liquids, a decrease in it most intensely occurs at comparatively low pressures (are less than 1000-1200 kgf/cm², Fig. 9a), whereupon during small changes in the pressure of liquid (to 600-700 kgf/cm²) a relative change in the volume (the volume strain of liquid) $\Delta V/V_0$ in the process of compression can be accepted proportional to a change in the pressure Δp, i.e., it approximately is subordinated to Hooke's law

$$\Delta p = \frac{\Delta V}{V_0} E = \frac{\Delta V}{V_0} \cdot \frac{1}{\beta}.$$

On the average during a change of the pressure from 1 to 1000 kgf/cm², the coefficient of the compressibility of mineral oils in the isothermal mode/conditions of compression decreases by 30-40o/o, and of the synthetic liquids of - 60-70o/o of its initial value (at atmospheric pressure and normal temperature). At high pressure (is more than 2500 kgf/cm²) a further pressure increase is not accompanied by a noticeable decrease in the volume of liquid, and consequently, an increase in the compression work of liquid with a pressure increase decreases and can become negligibly small.

The latter visually is confirmed by the empirical curves of the compressibility of silicone liquid, given in Fig. 9b.

In the general case it is possible to consider that the value of bulk rigidity modulus E (at t = 20°C and atmospheric pressure) for the mineral oils, utilized in hydraulic systems, is located of 13 500-17 500 kgf/cm². The lower limit of the corrected values of module/modulus E = 13 500 kgf/cm² corresponds to the widespread in aviation hydraulic systems oil of the AMG-10 of low viscosity, upper limit E = 17 500 kgf/cm² - to the more viscous oils (for example turbine), used in the hydraulic systems of other machines (presses, etc.).

In the widespread in hydraulic systems pressure range (0-500 kgf/cm²) value E in the function of pressure p virtually is subordinated to empirical regularity

$$E = E_0 + Ap,$$

where E_0 is elastic modulus at atmospheric pressure; A - the parameter, depending on the type of liquid and its temperature.

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Table 1. Parameter Ai module/modulus E_0.

(1) Масло	A				$E_0 \cdot 10^{-4}$ (2) в $кГ/см^2$			
	20° C	40° C	60° C	80° C	20° C	40° C	60° C	80° C
(3) Турбинное Л (ГОСТ 8675—62) . .	12,00	12,00	12,00	12,00	1,95	1,81	1,66	1,50
(4) АМГ-10 (ГОСТ 6794—53)	12,75	12,75	11,90	10,60	1,67	1,49	1,33	1,19

Key: (1). Oil. (2). in kgf/cm². (3). Turbine 1 (GOST [All-union State Standard] 8675-62). (4). AMG-10 (GOST 6794-53).

The values of parameter A and of module/modulus E_0 at the different temperatures for two used in hydraulic systems oils are given in Table 1.

Taking into account the change of the elastic modulus, which proceeds during a pressure increase, the average value of the coefficient of compressibility β the oil of AMG-10 for the widespread in hydraulic systems pressure range from 0 to 200 kgf/cm² and temperature t = 20°C

$$\beta = 7 \cdot 10^{-5}\ cm^2/kg.$$

For a comparison it is appropriate to indicate that elastic modulus they will become E = 2 x 10^6 kgf/cm², i.e., it is more than 100 times more the modulus of elasticity of mineral liquid.

The high indices of compressibility (by low elastic modulus) possess synthetic and, in particular, the ethylpolysiloxane liquids, the coefficient of compressibility of which (see Fig. 9b) by approximately 50o/o higher than the coefficient of the

compressibility of the liquids of mineral origin. For synthetic liquids the value of bulk rigidity modulus is located (8-10) 10^3 kgf/cm².

For water and the widespread working fluids on water principle (water-glycol, etc.) the average value of elastic modulus at relatively small pressures (to 200 kgf/cm²) can be taken as $2 \cdot 10^4$ kgf/cm².

During rapid (instantaneous) changes in the compression pressure, of liquid occurs on adiabatic curve, whereupon adiabatic elastic modulus higher than isothermal. In view of this during calculations of dynamic processes hydraulic drive is utilized not the isothermal, but adiabatic (dynamic) bulk modulus of the liquid of the aaaa which characterizes the compressibility the latter in the quickflowing processes change in the pressure and the absence of significant heat exchange between the liquid and the environment. The value of this bulk modulus is defined as product of the volume of liquid to pressure derivative in terms of the volume of where V and p - volume and the pressure of liquid.

$$E_a = V \frac{dp}{dV}, \qquad (6)$$

The adiabatic bulk modulus of the liquid of E_a usually is determined in practice by the acoustic method, based on the measurement of velocity a of the propagation of acoustic wave in this liquid by density ρ

$$a=\sqrt{\frac{E_a}{\rho}}.$$

In accordance with this, the module/modulus is defined as product of the density of liquid to the square of the speed of sound in it in the assigned conditions (pressure and temperature):

$$E_a=\rho a^2.$$

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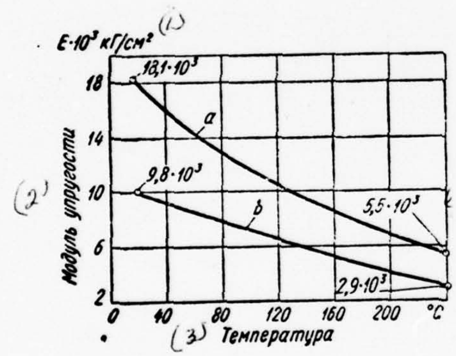

Fig. 10. Dependence of bulk modulus on the temperature: the curve of a) mineral oil; the curve of b) synthetic liquid.

Key: (1). kgf/cm². (2). Elastic modulus. (3). Temperature.

With an increase in the pressure on dp, the density of liquid is raised by $d\rho$ and becomes equal to

$$\rho = \rho_0 + d\rho,$$

where ρ_0 - the initial density at atmospheric pressure.

Taking into account that the mass of liquid $m = V\rho$, we have

$$\frac{d\rho}{\rho_0} = \frac{dV}{V_0};$$

in accordance with this adiabatic (dynamic) bulk modulus

$$E_a = \rho_0 \frac{dp}{d\rho}.$$

The dependence of the density of liquid on bulk modulus will be expressed

$$\rho=\rho_0+\rho\frac{dp}{E_a}.$$

The analysis of experimental data shows that during the calculation of the quickflowing processes in hydraulic systems (for example during calculations of the dynamic characteristics of hydroaggregates) adiabatic module/modulus for the used in hydraulic systems oils and a pressure range 50-200 kgf/cm² it is possible to accept $E_a \approx$ 1.15E, where E is a bulk modulus with of isothermic mode/conditions of compression.

Temperature effect. With an increase in the temperature, the bulk rigidity modulus of the working fluids of hydraulic systems decreases, in accordance with which the compressibility of these liquids with an increase in the temperature is raised, whereupon the

compressibility of more viscous oils is higher than the compressibility of less viscous oils of the same type. On the average the bulk modulus of the majority of working mineral fluids at atmospheric pressure and to temperature of 40°C is equal to (17-18) 10^3 kgf/cm² and decreases with temperature of 200°C to (9-10) 10^3 kgf/cm². The modulus of elasticity of synthetic (silicone) liquids decreases with these conditions from $10 \cdot 10^3$ kgf/cm² to (4.2-4.5) 10^3 kgf/cm².

Comparative experimental data on the dependence of bulk rigidity modulus on the temperature at pressure 210 kgf/cm² for a mineral oil (is curve a) and synthetic (silicone) liquid (is curve b) are given in Fig. 10.

Effect of the undissolved air. In view of the fact that the compressibility of air (gas) many times is higher than the compressibility of liquids (modulus of elasticity of air is equal approximately to the value of its absolute pressure), the presence in the liquid of air bubbles considerably reduces its bulk rigidity modulus, in consequence of which is raised the compliance/pliability (sag) of the exit component/link of hydraulic engine under the action of external load (is reduced the hardness of hydraulic mechanism).

the undissolved air leads also to the delay of hydraulic system in the final adjustment of signals and to the loss by it of stability against auto-oscillations.

Let us examine the effect of the undissolved air on the bulk modulus of elasticity of the liquid, which contains the undissolved air. For the analysis of this dependence, let us assume that in the volume of the $V_{ж}$ of liquid, which is located under pressure p, is contained undissolved air in the volume of $V_{в}$.

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A change of the $\Delta V_{в}$ of the volume of the aaaa of air during the compression of liquid-air mixture in isothermal mode/conditions (pressure p is raised to p + Δp) let us find from the following relationship (we assume that during air compression is not dissolved in liquid):

$$\Delta V_{e} = V_{e} \frac{\Delta p}{p}. \quad (7)$$

Consequently, the bulk modulus of air of E_{e} during compression in isothermal mode/conditions is numerically equal to that pressure hearth by which it is located:

$$E_{e} = V_{e} \frac{\Delta p}{\Delta V_{e}} = p.$$

Analogously it is possible to show that the bulk modulus of air during compression in adiabatic and polytropic mode/conditions will

be determined by expressions

$$E_{\text{в}}' = kp; \quad E_{\text{в}}'' = np,$$

where k and n are an adiabatic index and polytropic curve.

Consequently, the value of the modulus of elasticity of air by the polytropic mode/conditions of compression exceeds the value of elastic modulus by isothermal mode/conditions in n once and by adiabatic - in k once.

Accepting, that the bulk rigidity modulus of the $E_{ж}$ of liquid is retained constant during a change in the pressure on Δp, which is virtually correct at the being applied pressures, we find on the basis of expression (6) of the $\Delta V_{ж}$ of a change in the volume of the $V_{ж}$ of the liquid phase of liquid-air mixture during a change in the pressure on Δp (with p to p + Δp):

$$\Delta V_{ж} = \frac{V_{ж}}{E_{ж}} \Delta p. \qquad (8)$$

A change in the ΔV_c of the vclume of liquid-air mixture during a charge in the pressure cn Δp (from p_0 to $p = p_0 + \Delta p$):

$$\Delta V_c = \Delta V_{ж} + \Delta V_в = \left(\frac{V_{ж}}{E_{ж}} + \frac{V_в}{p}\right)\Delta p = \\ = \frac{V_{ж}\,\Delta p}{E_{ж}}\left(1 + \frac{V_в}{V_{ж}}\cdot\frac{E_{ж}}{p}\right),$$

where $V_{ж}$ and $V_в = V_{0в}\frac{p_0}{p}$ - the volume of liquid and undissolved air at pressure p; $V_{0в}$ are the initial air volume at pressure p_0.

In accordance with this the bulk modulus of liquid-air mixture

$$E_c = V_c \frac{\Delta p}{\Delta V_c} = \frac{V_{ж} + V_в}{\Delta V_{ж} + \Delta V_в} \Delta p, \tag{9}$$

where $V_c = V_{ж} + V_в$ - the volume of liquid-air mixture at pressure p.

After solving together given equations (7) - (9), we will obtain after conversion and simplification approximation for determining the bulk rigidity modulus of the liquid-air mixture (given bulk rigidity modulus) of E_c during its compression in isothermal mode/conditions from p_0 to p:

$$E_c = E_{ж} \frac{1 + \frac{V_{0в}}{V_{ж}} \cdot \frac{p_0}{p}}{1 + \frac{V_{0в} p_0}{V_{ж} p^2 E_{ж}}} \tag{10}$$

cr

$$E_{c} = E_{ж} \frac{p}{\bar{V}_{в} p + \bar{V}_{ж} E_{ж}},$$

where $\bar{V}_{в} = \frac{V_{0в}}{V_{0в} + V_{0ж}}$ are the relative vclume of air;
$\bar{V}_{ж} = \frac{V_{0ж}}{V_{0ж} + V_{0в}}$ - the relative initial volume cf liquid.

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From the given equations it follows that a change in the modulus cf elasticity of liquid-air mixture (given modulus of elasticity of working medium) it will be determined with the previously accepted assumption in essence by the mcdulus cf elasticity of air.

Thermal conductivity and liquid specific heat.

For absorption and absorption and evacuaticn from the hydraulic

system of the heat, which is isolated in work, it is necessary that the working fluids possess the determined values of specific heat and thermal conductivity. Specifically, the evacuation of heat from the places of its formation/education in hydraulic system in many respects depends on the thermal conductivity of liquid.

The thermal conductivity of liquids is characterized by the amount of heat which passes per unit time past the unit of area of the layer of liquid as thickness in the unit of length. In accordance with this the unit of thermal conductivity W/ (m•K), kcal/ (m•h•°C) and cal/ (cm•s•°C).

For the majority of oil-products, the thermal conductivity is equal approximately (4.0-4.8) 10^{-4} cal/cm•s•°C, for water - (1.3-.16) 10^{-3} cal/ (cm•s•°C).

The specific heat of working fluids whose hearth is understood the amount of heat, necessary for heating the unit of mass on 1°C, determines the intensity of an increase in the temperature in hydraulic system.

For the widespread working fluids of mineral origin, specific heat in temperature range 0-100°C on the average is equal to 0.45 kcal/ (kg•°C).

Pressure of saturated steams of liquids.

The pressure of saturated the pair of liquid is called the establish/installed in the closed space pressure pair, that is found at the given temperature in equilibrium with liquid. The pressure of saturated the pair of liquid must be known during the solution to the question concerning the suitability of the datum of liquid for a work at high temperatures, and also to evaluate cavitation characteristics of hydraulic system.

The evaporation of liquid occurs both from the surface and by formation of bubbles the pair (boilings) of liquid in all its volume, whereupon unlike the surface evaporation of liquid which occurs at any temperature, boiling liquid occurs only at the determined temperatures at which pressure the pair exceeds external pressure.

This pressure causes the true cavitation which begins when external pressure becomes below the pressure of saturated pair. At in the external pressure increase the boiling point it increases, at fall decreases, whereupon the intensity of the pressure buildup of pressure of vapor the higher, the higher its temperature.

The pressure of saturated the pair of homogeneous liquids (for example water) has for each temperature the determined value. But if in liquid dissolved any substance, then as a result of the interaction of the molecules of this substance and solvent is impeded the evaporation of the latter. In view of this unlike the homogeneous liquid which will boil at constant for this pressure temperature, the boiling point of complex liquids (mineral oils, etc.) at this pressure with the boil-off of light constituents is raised, but the pressure of saturated steams at the given temperature is reduced.

Fig. 11. Pressure of saturated steams of the widespread mineral oils.

Key: (1). mm Hg. (2). Pressure of saturated steams. (3). Industrial (4). Temperature of oil.

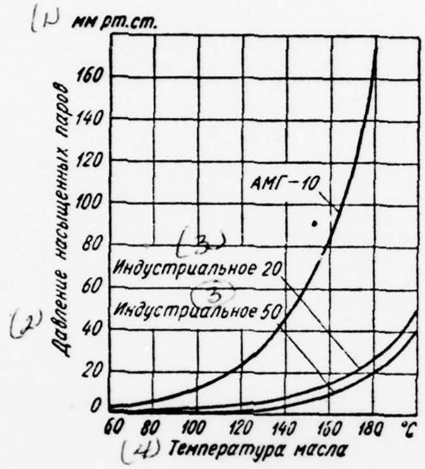

Figure 11 gives the values of the pressure of saturated steams of some types of the mineral oils, which are applied in the hydraulic systems of machines.

Cavitation of liquids.

With the question examined above concerning the pressure of saturated steams of liquid, is connected the phenomenon, obtain in practice the name of cavitation hearth by which is understood the local isolation from liquid in the zones of the reduced pressure of the vapors of liquid and gases (effervescence of liquid) with the subsequent failure (condensation) steam and the joining of gas bubbles with their incidence/impingement into the zone of elevated pressure.

The failure of bubbles is accompanied by the local hydraulic microblows of large recurrence.

Cavitation disrupts the normal mode of the operation of hydraulic system, and sometimes it exerts destructive action on its

aggregate/units. The destructive effect of cavitation undergo pumps, valves, valves and other hydroaggregates. With the advent of a cavitation in pumps, is reduced their feed, and also are observed high-frequency fluctuations of pressure in the pressure line of pump and impact loads on the parts, which are powerful to cause their premature output/yield from system. Furthermore, occurs the cavitation failure (corrosion) of the surfaces of parts with the formation/education on them of characteristic ulceration/pittings (surface porosity).

Up to now there is no strictly substantiated descriptions of the mechanism of the cavitation failure of the parts of hydroaggregates. Is most widely common the hypothesis, according to which this failure in essence occurs as a result high by the recurrence of the local hydraulic and thermal shocks, caused by the collision of particles of the liquid and by the compression of gas at the torque/moment of the joining of steam-gas bubbles, which are found in direct nearness of the walls of channel (enclosing surfaces).

Schematically the mechanism of the emergence of cavitation and its destructive effect is reduced to the following. With the appropriate decompression at any point of fluid flow, boils up

(occurs its discontinuity), the isolated in this case bubbles the pair and of gas they are carried along by flow they are transferred to the range of higher pressure in which the steam bubbles are condensed, and gas are compressed. Since the process of the condensation steam and the compression gas bubbles occurs instantly, the particles of liquid are moved to their centers at a high speed, which reaches according to the calculations of several hundreds of meters with second, as a result the kinetic energy of the colliding particles causes into the torque/moment of the joining of bubbles the local hydraulic microblows, which are accompanied by the excesses of pressure and temperature in the centers of bubbles (according to the calculations of the temperature in place the joinings of bubble can reach values of 1000-1500°C).

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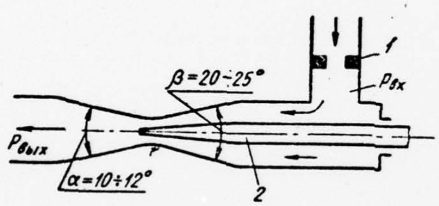

Fig. 12.

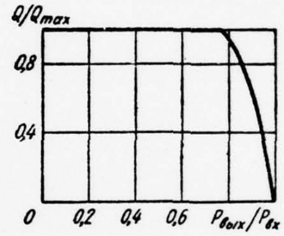

Fig. 13

Fig. 12. Diagram of cavitation device for the stabilization of fluid flow rate. Fig. 13. Curve, that characterizes the stability of expenditure/consumption through the cavitation device.

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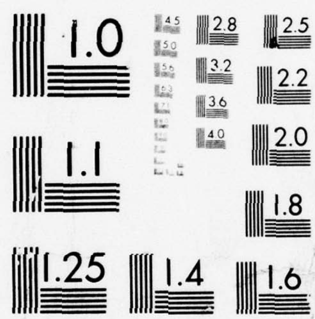

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in such a case, when the processes of cavitation flow/last near the walls of the enclosing channels, then the latter will undergo continuous thermal and hydraulic impacts (bombardments), which cause the local failures of walls. Under the action of the high temperatures in the presence of atmospheric oxygen, occurs also the active oxidation (corrosion) enclosing surfaces. The indicated percussion of particles of the liquid is supplemented by effect on the metal of electrolytic processes.

Methods of fight with cavitation and its consequences. The basic method of fight with cavitation is the maximum reduction in the evacuation/rarefaction in the zones of the possible cavitation, which partially can be reached by an increase in the ambient pressure. Specifically, the fight with cavitation in the suction barrel in essence is conducted by provision during the suction of this pressure which capable to overcome without the discontinuity of fluid flow hydraulic losses in suction line, in very inlet chamber, including also the friction, caused by inertia of liquid in these main lines.

For a reduction in the destructive effect on the part of the aggregate/units of cavitation, are applied stable against corrosion materials (steels with the addition of chromium and nickel) and the

coatings of the parts, washed by the cavitated liquid. As a rule, the durability of materials against cavitation failure is raised with an increase in their mechanical strength (hardness) and in the chemical (oxidizing) durability, the better/best results giving the materials in which are combined both these qualities. Least stable are the cast iron and the carbon steel, more stable they is bronze and the stainless steel. Most stable of the known materials is titanium.

Practical use of a cavitation. Cavitation frequently is utilized for practical target/purposes. Specifically, it is utilized in devices (Fig. 12) for the stabilization of fluid flow rate. Device consists of throttle washer 1, which gauges consumption of liquid, and axisymmetric choke needle 2, employed for the introduction of device to cavitation operating mode.

With decompression of $p_{вых}$ at output/yield, in this case from venturi nozzle, at the constant pressure of $p_{вх}$ at the inlet into it the speed of fluid flow is raised, in accordance with which pressure in the narrowed section of nozzle is reduced. After a reduction in this pressure to the value, which corresponds the cavitation onset of the liquid, the latter boils up. Since the friction of nozzle after this increases proportional to the

cavitation intensity, which, in turn, is raised with an increase in the pressure differential, the nozzle flow after the emergence of cavitation is stabilized, being retained constant independent of a further decompression at the nozzle outlet. During a decrease in this pressure, only is expanded the zone of cavitaticn on diffuse part, beginning from the narrowed section.

Similar cavitation devices (nozzle) utilize for the stabilization of expenditure/consumption during fluctuations of pressure at output/yield. They provide the flcw-rate control of liquid in the large range of the relaticns cf the expenditure/consumptions ($\geqslant$10) during the simultaneous stabilization cf expenditure/consumption in each mode/conditicns.

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Figure 13 shows curve the dependence cf the adjustable expenditure/consumption Q of liquid on jump/drop in the pressure on nozzle at the different values (from 10 tc 30 kgf/cm²) of the pressure of $p_{вх}$ at inlet and the pressure of the $p_{вых}$ at output/yield, which varies from 0 to the input (30 kgf/cm²).

Measurements are carried out with fluid flow rate from 500 to 40 000 cm^3/s at the pressures of p_{ex} = 10; 20; 25 and 30 kgf/cm^2.

From curve/graph it follows that the fluid flow rate was retained constant (coefficient of expenditure/consumption μ varied from 0.96 to 0.97) over a wide range of mode/conditions. The disturbance/breakdown of the stabilized expenditure/consumption occurs virtually at the values of critical pressure at the output/yield of the $p_{eux} \approx p_{ex}$, where the p_{ex} - the pressure of flow at the nozzle entry.

Cavitation effect widely is utilized for cleaning welding scale and different contaminators from articles, and also for the surface treatment of parts (removal/taking from the articles of scale and projecting edges, improvement in the surfaces, etc.) which is reached because of cavitation erosion.

Being applied liquids.

In the hydraulic systems of machines, are commonly used the

working fluids of mineral origin with kinematic viscosity in 50°C approximately 10-150 c. st. In the hydraulic systems of the machines, intended for a work under stable temperature conditions at pressures is less than 100 kgf/cm², are commonly used oils with viscosity 20-40 c. st. (at 50°C), and at pressure to 200 kgf/cm², they are commonly used with viscosity 40-60 c. st. In the hydraulic systems of presses with pressure 500-600 kgf/cm² viscosity the liquids frequently lead to 110-150 c. st.

The hydraulic systems of many machines and installations work at high temperatures, which reach 300°C and above with which mineral liquids and their mixture are not suitable for a work. Better/best from this viewpoint the existing mineral liquids are suitable for a work at temperature not above 150°C. At more high temperatures mineral liquids enter into a reaction with atmospheric oxygen and are decompose/expanded with the isolation of solid films and tarry residue/settlings, which are powerful to upset the operation of hydraulic system. Furthermore, an increase in the temperature is accompanied by an increase in the pressure of saturated steams of liquid, which contributes to the emergence of cavitation.

In view of this for a work at high temperatures (150°C and

above) it is possible to apply without the special coolers only the high-temperature liquids, most widely accepted from which are the synthetic liquids, in particular polysiloxane and silicon, which combine in themselves both high-temperature and low-temperature properties. In practice are common the polysiloxane (silicone) liquids, which possess satisfactory temperature-viscous characteristics in wide temperature range and differ in terms of mechanical strength and stability against oxidation, and also in terms of high thermal stability, retaining it even during heating in the presence of atmospheric oxygen. In contact with air, they allow/assume prolonged heating with temperatures to 250°C, in enclosed systems without air inlet of their it is possible to long utilize at temperature to 370°C.

Simultaneously these liquids allow/assume the work of hydraulic system at temperatures - 60°C and below.

Furthermore, they possess also the low pressure of saturated steams value of which at temperature of 60°C it does not exceed 1 mm Hg, or they are virtually fire-resistant and preventing the propagation of fire/light.

Deficiency/lacks in the polysiloxane liquids include the fact that they dissolve all the existing plasticizers of synthetic rubbers. Therefore the ferrules, prepared from these rubbers, become for brief operating time brittle and are cracked, as a result of which hydroaggregates unavoidably they lose airtightness.

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Furthermore, polysiloxane liquids are inferior to mineral oils according to the antiabrasive and lubricating qualities.

Sometimes of applying hydraulic systems of temperature, they reach such high values, that is eliminated the possibility of applying both mineral and existing synthetic liquids. In view of the fact that the temperature of the work of hydraulic systems continuously are raised, promising is the application/use as the working fluids of the hydraulic systems of liquid metals with low melting point which are utilized at present as heat-transfer agents in nuclear reactors [7].

Most promising is the eutectic alloy, which consists of 77o/o of sodium even 23o/o of potassium, which is the silver metal, similar to mercury. The point of its melting (eutectic point) is equal - 12°C and boiling (at atmospheric pressure) 850°C. Bulk modulus at temperature of 40° is equal to 52 500 kgf/cm² [7].

FLOW OF LIQUIDS ON CONDUIT/MANIFOLDS HYDRAULIC SYSTEMS.

Flow of liquid along the conduit/manifolds of hydraulic system and the channels of its aggregate/units is accompanied by hydraulic losses (losses of pressure and energy) whose values depend, other conditions being equal, on flow conditions, and also under specific conditions on form, size/dimensions and the roughness of conduit/manifold.

From course "hydraulics" it is known that they distinguish two flow conditions of liquid in conduit/manifolds [19]: by laminar, characteristic laminar flow without the mixing of particles and

pulsations of velocity, and turbulent, that is accompanied by the intense mixing of particles of the liquid and by the pulsations of velocities. Transition from the laminar to turbulent mode/conditions begins under certain conditions, characterized by Reynolds' number (criterion) by Re, which are the dimensionless quantity, which relates the average for section speed of fluid flow u, diameter d of the section of conduit/manifold (linear dimension of channel) and the kinematic modulus of viscosity of the liquid of ν.

In application to the motion of liquid in the conduit/manifolds of round cross-section, this criterion takes the form

$$Re = \frac{ud}{\nu}. \qquad (11)$$

In the conduit/manifolds (channels) of noncircular and ring cross-sections

$$\mathrm{Re} = \frac{4ru}{\nu},$$

where r = F/ - the hydraulic radius of the section of flow, which is the relation of area F of the section of flow to wetted perimeter χ.

For a ring duct (slot)

$$r = \frac{r_2 - r_1}{2},$$

where r_1 and r_2 are external and inside radiuses of slot.

To the torque/moment of the transition of stream-line conditions to turbulent and back correspond in these conditions determined Re: with Re less than critical - flow laminar, with Re more critical -

turbulent. To the stream-line conditions of flow of liquid in the hydraulically smooth metal tubes of round cross-section corresponds to Re $\leqslant$ 2200-2300, turbulent - Re $\geqslant$ 2200-2300.

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Calculation of losses of pressure during flow of liquid in conduit/manifold.

The stream-line conditions of flow. The loss of pressure Δp on the cylindrical linear segment of conduit/manifold, caused by the frictional resistance of liquid, is calculated during the stream-line conditions of flow (Re < 2300) from the known formulas, obtained from

the equation of Poiseuille [19]:

$$\Delta p = p_1 - p_2 = 32\,\frac{\mu L u}{d^2};$$

$$\Delta p = \frac{128}{\pi}\cdot\frac{L}{d^4}\,\mu Q = \frac{128}{\pi}\,\nu\rho\,\frac{L}{d^4}\,Q = R_A Q, \qquad (12)$$

where p_1 and p_2 are pressures in the beginning and at the end of the cut of conduit/manifold; μ and ν are dynamic and kinematic viscosity of liquid; ρ - the density of liquid; L and d - length and the diameter of the internal section of the cut of conduit/manifold in question; Q and u - the average values of expenditure/consumption and velocity of liquid in conduit/manifold; $R_A = \frac{128}{\pi}\,\nu\rho\,\frac{L}{d^4}$ — are hydraulic plumbing friction during laminar flow of liquid.

The given equations are valid for the conduit/manifolds of this length at which it is possible to disregard velocity pressure (entry loss) in comparison with losses of pressure on friction.

In such a case, when the unknown value is the fluid flow rate or the diameter of the section of conduit/manifold, more preferable to use the expressions, obtained by the transformation of expressions (12):

$$Q=\frac{\pi d^4 \Delta p}{128\mu L}=\frac{\pi}{128}\cdot\frac{1}{\nu\rho}\cdot\frac{d^4}{L}\Delta p. \tag{13}$$

After introducing coefficient $\lambda=\frac{64}{\mathrm{Re}}=\frac{64\nu}{ud}$ and after producing the appropriate transformations, expression (12) it is possible to lead to form

$$\Delta p=\frac{64}{\frac{ud}{\nu}}\cdot\frac{L}{d}\cdot\frac{u^2}{2}\rho=\lambda\frac{L}{d}\cdot\frac{u^2}{2}\rho=\lambda\frac{L}{d}\cdot\frac{\rho}{2}\cdot\frac{Q^2}{f^2}, \tag{14}$$

where λ = 64/Re - the dimensionless coefficient of friction drag of the stream-line conditions of flow; f is a section of

conduit/manifold.

In certain cases it is represented advisable to determine loss of head in the units of length

$$H=\frac{\Delta p}{\rho g},$$

or taking into account the preceding/previous expression

$$H=\lambda\frac{L}{d}\cdot\frac{u^2}{2g}=\lambda\frac{L}{d}\cdot\frac{Q^2}{2gf^2}. \qquad (15)$$

During flow of liquid in conduit/manifolds, appears also the supplementary resistance, caused by contraction and the other distortions of the cylindricity of the section of duct, and also by

cooling the layers of liquid (being contacted with the walls of duct), that are accompanied by an increase in the viscosity of these layers as compared with the average viscosity over the section of the duct which enters into the calculation during the determination of viscosity. Taking into account these factors λ during practical calculations of ducts, one should calculate for the stream-line conditions of the flow:

$$\lambda = \frac{75}{Re}.$$

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Turbulent flow conditions. Turbulent flow conditions of liquid in the channel (conduit/manifold) of constant section is accompanied by the dissipation of kinetic energy as a result of the random movement of particles of the liquid.

The losses of pressure in conduit/manifold during stationary turbulent flow (Re > 2300) are determined from formula [see also formula (14)]

$$\Delta p = p_1 - p_2 = \lambda_T \frac{L}{d} \cdot \frac{Q^2 \rho}{2f^2}, \qquad (16)$$

where λ_T - loss factor of turbulent flow, depending on Re and the relative roughness of the internal surface of conduit/manifold; for a hydraulically smooth conduit/manifold this coefficient usually is calculated for conditions 2300 < Re < 8000 for the semi-rational formula of Blasius [6]

$$\lambda_T = 0{,}3164\,\mathrm{Re}^{-0,25}. \qquad (17)$$

By hydraulically smooth pipe it is accepted to count this duct in which the projections (roughness) are hidden in thicker than the laminar boundary layer of liquid of walls. In view of the fact that with increase in Re the thickness of this layer decreases, the projections of the roughness of duct with known Re can be bared, as a result the duct will stop to be hydraulically smooth, and to the coefficient of the losses of λ_T will have an effect the surface roughness of the walls of conduit/manifold [7].

Drawn tubes of steel, brass and copper it is possible to assume/take hydraulically smooth on all range Re, possible in the hydraulic systems in question.

For smooth drawn tubes it is possible with sufficient accuracy/precision to assume/take during practical calculations

$$\lambda_T = 0{,}025.$$

After designating by

$$R_T = \lambda_T \frac{L\rho}{2df^2},$$

formula (16) let us rewrite in general form

$$\Delta p = R_T Q^2,$$

where R_T is hydraulic plumbing friction during turbulent flow conditions.

Hydraulic conductivity.

In practice during calculations and investigations, frequently they use the concept of the hydraulic conductivity K whose hearth understands the value, reciprocal to the hydraulic resistance R:

$$K=\frac{1}{R}. \quad (18)$$

For the stream-line conditions of flow in conduit/manifolds, hydraulic conductivity is expressed according to equation (12)

$$K_{A}=\frac{1}{R_{A}}=\frac{Q}{\Delta p};\quad Q=K_{A}\Delta p.$$

For turbulent flow conditions, hydraulic conductivity is expressed

$$K_T = \frac{1}{R_T} = \frac{Q^2}{\Delta p}; \quad Q = \sqrt{K_T \Delta p}.$$

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Local hydraulic losses.

The local hydraulic losses are called pressure drops across the overcoming of resistance during flow of liquid through the cell/elements of hydroaggregates and accessories (through the local resistance).

Losses of pressure in the local resistance are expressed in the portion/fractions of velocity pressure and are computed in the general case according to formulas [6]

$$\Delta p = p_1 - p_2 = \zeta \frac{u^2\rho}{2}; \quad \Delta H = \frac{\Delta p}{\rho g} = \zeta \frac{u^2}{2g}, \qquad (19)$$

where Δp and ΔH - the loss of pressure and pressure head; ζ is a coefficient of the local resistance; $u^2\rho/2$ and $u^2/2g$ are velocity pressure and velocity head of flow; here u is the average over the section of flow velocity of liquid at output/yield from resistance.

Coefficient

$$\zeta = \frac{\Delta H}{u^2/2g} = \frac{\Delta p}{u^2\rho/2},$$

being the ratio of pressure head ΔH to velocity head $u^2/2g$, or losses of pressure Δp to velocity pressure $u^2\rho/2$, shows, what part of velocity head or velocity pressure is expend/consumed on the overcoming of this local resistance.

During calculations of hydraulic systems, they use the experimental data on coefficients ζ, determined by means of the spills of concrete/specific/actual hydroaggregates [7].

Sudden expansion and the contraction of channel (conduit/manifold). For practical calculations of hydraulic systems great significance they have calculations of losses with sudden expansion and the contraction of conduit/manifolds. Specifically, the conditions, which correspond to the sudden expansion of channel, occur during the input/introduction of liquid from duct into

actuating cylinders, pneudraulic storage battery/accumulators, filters and other tanks.

During the sudden expansion of channel (Fig. 14) the flow of liquid is characterized by the gradual expansion of liquid jet, which is accompanied by deceleration of its flow and by the loss of pressure and energy, which proceeds in essence as a result of vortex formation after having emerged of jet from the draft of the channel.

The loss of pressure head with the sudden expansion of conduit/manifold is equal to velocity head, calculated from the lost velocity (borda - Karno's theorem):

$$\Delta H = \frac{(u_1 - u_2)^2}{2g}; \quad \Delta p = \Delta H \rho g,$$

where u_1 and u_2 are the average speed in the conduit/manifold of small (before expansion) and large (after expansion) cross sections.

Taking into account the equation of the constancy of expenditure/consumptions ($u_1 f_1 = u_2 f_2$) the last/latter equation it is possible to present in the form

$$\Delta H = \frac{u_1^2}{2g}\left(1 - \frac{f_1}{f_2}\right)^2$$

or

$$\Delta p = \frac{\rho}{2}(u_1 - u_2)^2 = \frac{\rho u_1^2}{2}\left(1 - \frac{f_1}{f_2}\right)^2,$$

where $f_1 = \pi d^2{}_1/4$ - the section of the conduit/manifold of the small diameter d_1 (before expansion); $f_2 = \pi d^2{}_2$ - the section of large-diameter conduit/manifold D_2 (after expansion.

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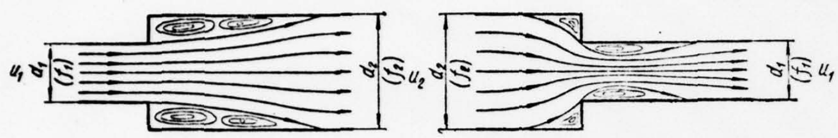

Fig. 14.

Fig. 15.

Fig. 14. Sudden expansion of channel.

Fig. 15. Sudden contraction of channel.

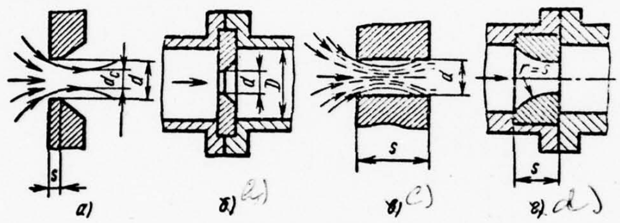

Fig. 16. Forms of throttle channels.

During discharge from the conduit/manifold of a small section into conduit/manifold with relatively larger cross section (cylinder, etc.) u_2 is practically equal to zero, in accordance with which the loss of head and pressure

$$\Delta H = \frac{u_1^2}{2g} \quad \text{or} \quad \Delta p = \frac{\rho u_1^2}{2}.$$

With the sudden contraction of channel (Fig. 15) hydraulic losses are caused in essence by friction and strain (compression) of flow upon the entrance into narrow channel and partially by vortex formation during flow expansion after its compression.

The loss of head and pressure with the contraction of channel

$$\Delta H = \zeta \frac{u_1^2}{2g}; \quad \Delta p = \zeta \frac{u_1^2 \rho}{2},$$

where ζ is a drag coefficient (with the large contraction of $\zeta_A =$ C.5).

Discharge through the opening/aperture in fine/thin wall. In hydroaggregates, and also in measuring meters is common the local resistance in the form of opening/aperture (or slot) in fine/thin wall.

By fine/thin is understood this wall, with which the discharging jet is contacted only with the sharp edge of opening/aperture, turned inside container, and it clear the lateral surface of opening/aperture, which corresponds to the complete compression of jet [6]. Experience/experiment shows that the length of the section on which occurs the compression of jet, can be under specific conditions equal to 0.5 diameters of opening/aperture, and

consequently, in order to avoid the contact of jet against the surface of opening/aperture, thickness s of wall (length of opening/aperture) it must be not more than its diameter d.

The length of opening/aperture s can be decreased without the disturbance/breakdown in this case of the hardness of wall down to any small path length of the execution of the edge of opening/aperture by the schematic, given in Fig. 16a.

From course "hydraulics" [19] is known that jump/drop in the pressures Δp and expenditure/consumption Q of the liquid through the opening/aperture in question are connected by equation

$$Q = \mu f u_T = \mu f \sqrt{\frac{2\,\Delta p}{\rho}}, \tag{20}$$

where μ is a coefficient of expenditure/consumption; f - the area of the section of opening/aperture; $u_T = \sqrt{\frac{2\,\Delta p}{\rho}}$ are the calculated (is theoretical) speed of fluid flow; ρ - the density of liquid.

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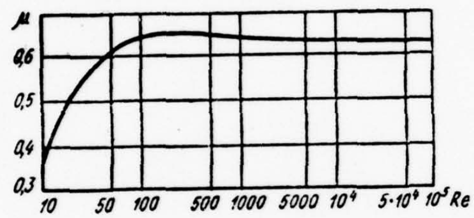

Fig. 17. Dependence of the coefficient of the expenditure/consumption for a round opening/aperture in fine/thin wall with sharp edges on Re.

The coefficient of expenditure/consumption is equal to the product of the $\mu = \varphi\varepsilon$, where ϕ and ε - coefficients with respect to speed and the compression of jet in its output/yield from opening/aperture.

Velocity coefficient:

$$\varphi = \frac{u}{u_T},$$

where $u < u_T$ — are actual speed of liquid taking into account braking layers of the edge of opening/aperture; $u_T = \sqrt{2gH}$ or $u_T = \sqrt{\frac{2\Delta p}{\rho}}$ — are the ideal exhaust velocity of ideal fluid.

During the escape of the low-viscosity liquids through the round opening/aperture in fine/thin wall with sharp edges it is possible to assume/take ϕ = 0.97-0.98.

The contraction coefficient of jet

$$\varepsilon=\frac{f_c}{f},$$

where $f_c<f$ - sectional area flowing its narrow (compressed) place; f is a sectional area of opening/aperture.

The compression of jet is caused by the fact that the particles of liquid approach the opening/aperture along curved paths. Because of this the jet during discharge from opening/aperture blows away at input sharp edge from wall and at certain distance from it is compressed, as a result of which the sectional area of jet in its narrow section is less than the sectional area of opening/aperture.

During the turbulent mode/conditions of the escape of low-viscosity liquids (ν = 10-60 c. st.; Re > 50) from the round opening/aperture of small section the contraction coefficient of jet it is possible to accept for the approximate computations by constant: $\varepsilon=0{,}64$.

In accordance with this the coefficient of expenditure/consumption for these widespread values φ and an ε it is possible to accept equal $\mu = \varphi\varepsilon \approx 0,62$.

This value it can be recommended during practical calculations of the widespread mode/conditions of the discharge through opening/apertures in fine/thin wall with sharp edges. The precise value of the coefficient of expenditure/consumption μ for concrete/specific/actual opening/aperture and Re is determined by spills from formula

$$\mu = \frac{Q}{Q_T},$$

where Q - the measured fluid flow rate through the opening/aperture; Q_T are the theoretical flow rate, calculated according to expression

$$Q_T = u_T f = f \sqrt{\frac{2\,\Delta p}{\rho}}.$$

Figure 17 gives the experimental curve/graph of the dependence of the coefficient of expenditure/consumption μ for a round opening/aperture with sharp edges on Re. The observing here sharp increase in the coefficient μ of small Re (Re < 100) is caused by growth of velocity coefficient ϕ. In this zone Re, the role of viscosity is so great and braking the speed of flow of the edges of opening/aperture so considerably, that the compression of jet practically is absent $(\varepsilon \approx 1)$, in accordance with which the coefficient μ is raised approximately proportional to increase of Re. Certain fall μ, beginning with Re > 500 to Re = 10^5, is caused by decrease with these Re of contraction coefficient ε.

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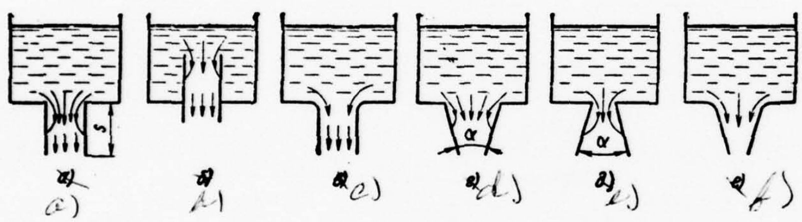

Fig. 18. Schematics of nozzles.

During executicn on the entering edges of facets or roundings with the relative depth $h/d = 0.2-0.4$, where d and h - diameter and the height/altitude of facet, the coefficient cf expenditure/consumption of $Re > 100$ is raised tc $\mu \approx 0.8$. With the rounding of entering edges with a radius cf $r \approx d$ even $50 < Re < 2 \cdot 10^4$ the compression of jet practically is removed, in view of which the coefficient of expenditure/consumpticn is raised to $\mu \approx 0.95$.

The blunting (rounding) of the edges cf opening/aperture can occur as a result of their wear in operation.

The coefficient of expenditure/consumpticn through the diaphragm actually does not depend on that, does occur discharge from opening/aperture into the atmosphere (unflooded opening/aperture) or into the space, filled by liquid (flooded opening/aperture) under atmospheric pressure.

The given value of coefficient μ is correct only for the ideal compression of jet, which occurs when opening/aperture is located on this distance from side walls of contairer (conduit/manifold) that the latter do not affect the character cf the formation of jet and,

consequently, also on the character of discharge. Practically the compression of jet will be ideal, if distance from the walls of container to opening/aperture the not less triple diameter of opening/aperture. With the less distance of wall, partially guide liquid jet with approach to opening/aperture, in view of which it is compressed to a lesser degree, than during discharge from the reservoir of the unlimited size/dimensions. As a result of this, the contraction coefficient and, consequently, also the coefficient of expenditure/consumption are raised.

For the widespread in practice case of the installation of throttle diaphragm in the duct of flow-meter device (see Fig. 16b) the fluid flow rate through the diaphragm

$$Q=\mu f\sqrt{\frac{2\Delta p}{\rho\left[1-\left(\frac{d}{D}\right)^4\right]}},$$

where D and d - the diameters of the section of duct and choke opening/aperture in diaphragm; Δp and ρ - the pressure differential of liquid and its density.

Flow of liquid through nozzles. Cap/fillings are called stub duals with the constant or being changed section along the length. Cap/fillings are applied in hydraulic systems, when it is required to ensure the required energy system performances or to shape according to the assigned law the jet of the escape/ensuing of nozzle liquid.

In practice are common the external cylindrical nozzles or the nozzles, which emerge from reservoir outside (Venturi's nozzle), shown in Fig. 18a and Fig. 16c. With $s/d > 2.5$-3 compression of jet at output/yield from nozzle is absent, i.e.: $\varepsilon = 1$ (jet concerns the trailing edges of opening/aperture), and consequently, the diameter of jet cross-sectional area is equal to the diameter of opening/aperture, in accordance with which $\mu = \phi$. However, in this case the speed of fluid flow somewhat decreases as a result of the action of viscous drag, in view of which the coefficient ϕ will be less than of the discharge through the opening/aperture in diaphragm; furthermore, the coefficient ϕ of this length of nozzle will somewhat depend on Re.

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Practically the values of coefficients μ and ϕ in the case of low-viscosity liquids can be assume/taken for nozzles equal to $\mu = \phi$ = 0.82. Consequently, fluid flow rate through the external cylindrical nozzle exceeds the expenditure/consumption through the opening/aperture of the same diameter in fine/thin wall by approximately 30o/o.

During an increase in the jump/drop in the nozzle pressure, the expenditure/consumption through it increases, and pressure in the contracted section of jet is reduced. However, upon reaching of certain jump/drop and respectively certain pressure by which appears the cavitation of liquid, expenditure/consumption through the nozzle is stabilized and a further increase in the pressure differential it does not produce an increase in the expenditure.

The role cap/filling in the hydraulic systems of machines usually makes opening/apertures in the thick walls of hydroaggregates (see Fig. 16c), if wall thickness is greater than diameter d of opening/aperture 2.5-3 times.

Examined/considered cylindrical nasadok (or respectively opening/aperture in the wall of the housing of hydroaggregate) can be improved by means of the rounding of entering edge (see Fig. 18c), whereupon with an increase in the rounding the coefficient of expenditure is raised. But if we outline nozzle on the duct of the surface of the jet, which escape/ensues into opening/aperture, then the compression of jet will come to the minimum. Similar nasadok, called taper, provides the coefficient of expenditure/consumption, the close to unity, and the stable operation of discharge.

As a result of complexity the executions of taper nozzle its configuration in practice replace by configuration on the friend of circle (see Fig. 16d), whereupon in the limit when radius r of the curvature of entering edge is equal to thickness s of wall, similar cylindrical nasadok practically is converted into taper.

The values of coefficients μ and ϕ in the case of the smooth rounding of entering edges can be assume/taken depending on Re equal to: $\mu = \phi = 0.98 - 0.96$, whereupon to higher Re correspond the

smaller values of coefficient μ and vice versa.

Are applied also the nozzles, which enter inside reservoir (Borda mouthpiece, see Fig. 18b). Flow of liquid in this nozzle is analogous with flow in external nozzle; however, the conditions of the entrance of liquid are somewhat deteriorated as a result of the large curvature of the bending of flow lines; therefore the coefficient of the expenditure/consumption of this nozzle less ($\mu \approx$ 0.72), than external nozzle.

In the hydraulic systems of machines, are applied both nozzles of Venturi and the Borda mouthpieces. Specifically, the Borda mouthpieces are applied in the partition devices of the tanks of the hydraulic systems in which for the elimination of the incidence/impingement into the partition branch of mechanical particles with high density is provided for certain unproduced volume of liquid.

Besides cylindrical external nozzles they are applied the conical convergent (converging nozzle sections, Fig. 18d) and divergent (exit cone/diffusers, Fig. 18e) cap/fillings. The

convergent cap/fillings, which ensure the minimum losses of pressure, will find use in hydraulic boosters of the type "jet pipe", and also in technology for the formation of fire and monitoring jets and etc.

From the viewpoint of the provision for the smallest losses of pressure and savings of power, the best results provide the taper convergent nozzles (see Fig. 16d and 18f). The coefficient of the expenditure/consumption of these nozzles depends on the angle α, with increase in which it is raised, reaching at α = 45° value μ = 0.96-0.98.

The divergent cap/fillings (see Fig. 18e) they are applied when it is required to obtain large expenditure/consumptions with small jump/drops in pressure and section of channel. Furthermore, they are applied, when it is required to convert the kinetic energy of flow into the energy of piezometric pressure, which is required, for example, in ejectors, exit cone/diffusers, etc.

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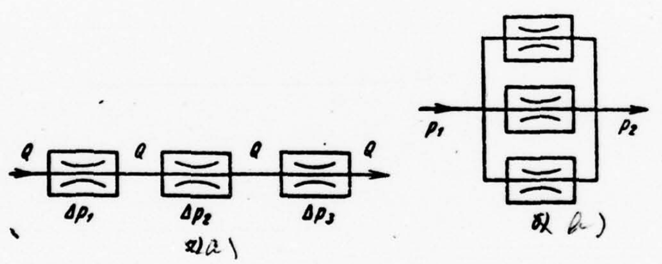

Fig. 19. Compounds of the local resistance: a) consecutive; b) parallel.

Angle of taper α cap/filling is usually equal to 14°. At higher angles is feasible the flow breakaway of liquid from the walls of nozzle, as a result of which the expenditure/consumption decreases.

Hydraulic conductivity of the local resistance. Fluid flow rate through the local hydraulic resistance frequently also is expressed as the hydraulic conductivity of flow area.

On the basis of equation (20) we can write (correction for a difference in the velocity pressures in control points we disregard)

$$Q = \frac{1}{R_м} \sqrt{\Delta p} = K_м \sqrt{\Delta p}, \qquad (21)$$

where $R_м = \frac{1}{K_м}$ - the hydraulic resistance of the local resistance; $K_м = \mu f \sqrt{\frac{2}{\rho}}$ are hydraulic conductivity of the local resistance (hydraulic element in question); $\mu = \frac{1}{\sqrt{\zeta}}$ and ζ are coefficients of expenditure/consumption and resistance.

Calculation of the compounds of the local resistance. With laminar flow the hydrobends are calculated by analogously to electrical circuits.

In the case of series connection and resistance, the fluid flow rate is retained on all sections by constant/invariable (Q = const), and loss of pressure on these sections they are summarized, i.e., the equivalent hydraulic resistance of several successive resistance of R_i equal to the sum of these resistance (Fig. 19a).

For the stream-line conditions of flow these losses

$$\Delta p_{экв} = \sum_{i=1}^{i=n} \Delta p_i,$$

where Δp_i - loss of pressure in one resistance.

A characteristic example of the tandem arrangement of resistance is the multiple-disk (package) throttle/choke (see Fig. 77b).

With turbulent flow taking into account the equality of $\Delta p_i = R_i Q^2$ the expression for losses of pressure during series connection n of resistance takes the form

$$\Delta p_{экв.т} = Q^2 R_{экв} = Q^2 \sum_{i=1}^{i=n} R_i,$$

where $R_{экв} = \sum_{i=1}^{i=n} R_i$ - equivalent hydraulic resistance.

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After introducing by analogy with equation (18) the concept of the equivalent hydraulic conductivity of $K_{экв} = \mu f \sqrt{\frac{2}{\rho}}$ a series of the series-connected resistance and taking into account that

$$\Delta p_i = \frac{Q^2}{K_i^2}$$

hydraulic losses for turbulent flow conditions can be presented in the form

$$\Delta p_{\text{экв. т}} = \sum_{i=1}^{i=n} \Delta p_i = \frac{Q^2}{K_{\text{экв}}^2},$$

where

$$\frac{1}{K_{\text{экв}}^2} = \frac{1}{K_1^2} + \frac{1}{K_2^2} + \frac{1}{K_3^2} + \cdots + \frac{1}{K_n^2}.$$

After using the last/latter dependence to package diaphragm (quadratic) throttle/choke (see Fig. 77b), we will obtain (losses a friction in the chambers of the housing of throttle/choke we disregard)

$$\frac{1}{K^2_{экв}} = \sum_{i=1}^{i=n} \frac{1}{K^2_i} = \frac{n}{K^2_i},$$

where n - the amount of throttling washers in package; $K_i = \mu f_i \sqrt{\frac{2}{\rho}}$ - the hydraulic conductivity of one washer; $K_{экв} = \mu f_{экв} \sqrt{\frac{2}{\rho}}$ - the equivalent hydraulic conductivity of the throttle/choke, which is of one washer the area of the $f_{экв}$, loss of pressure on which with the

same expenditure/consumption are equal to losses in package throttle/choke of several washers to the area of f_t each.

On the basis of resulting expressions

$$f_t = f_{\text{экв}}\sqrt{n}.$$

Consequently, the total area of the throttling opening/apertures in the washers of package throttle/choke will be with the same expenditure/consumption into the $\sqrt{n}$ of times more than the area of the opening/aperture of single-disk throttle/choke. For example, the area of opening/apertures in nine-disk throttle/choke will be 3 times more, under otherwise equal conditions, the area of the opening/aperture of single-disk throttle/choke.

Taking into account the given expenditure/consumption of package throttle/choke

$$Q = \mu_n f_t \sqrt{\frac{2}{\rho}}\sqrt{\Delta p},$$

where μ_n is an equivalent coefficient of the expenditure/consumption of package throttle/choke.

During parallel connection of hydraulic resistance (Fig. 19b) the pressure differentials, characterizing the hydraulic losses in branchings off, in each branch identical. Expenditure/consumptions through each of resistance, in each branch identical. Expenditure/consumptions through each of the resistance directly proportional to the area of choke channels or inversely proportional to their resistance, and the total expenditure/consumption through all the local resistance is composed of the expenditure/consumptions in separate branchings off (conductivities are added). This expenditure/consumption Q loss of pressure Δp with turbulent flow

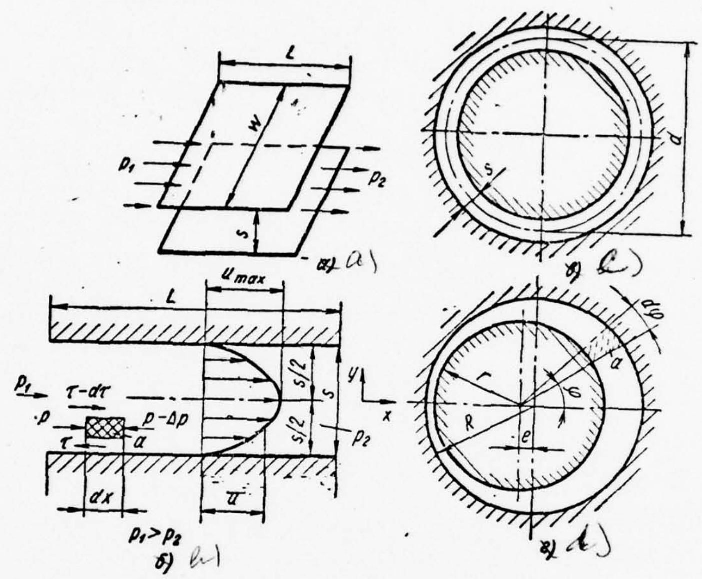
$$Q=\sum_{i=1}^{i=n} Q_i;$$

$$\Delta p = p_1 - p_2 = \text{const}.$$

Losses during flow in main lines. Expression for the calculation of the total hydraulic losses in the main line, which consists of conduit/manifolds and the local resistance, can be written in the form

$$\Delta p = \sum \zeta_i \frac{u^2\rho}{2} + \sum \lambda \frac{L}{d} \cdot \frac{u^2\rho}{2}.$$

Fig. 20. The design diagrams of flow of liquid in the capillary slots: a and b) by plane; c and d) circular.



After presenting the total drag coefficient of an entire main line in the form

$$\zeta_M = \sum \zeta_i + \sum \lambda \frac{L}{d},$$

we find the permissible average speed of flow u in main line taking this into account of the coefficient:

$$u = \sqrt{\frac{2\,\Delta p}{\rho \zeta_M}},$$

where Δp are the permissible (assigned by technical documentation) losses of pressure in main line.

Practically they allow/assume for the widespread hydraulic systems with operating pressure 150-200 kgf/cm² Δp = 2-6 kgf/cm²; u = 10 m/s (for pressure and drainage lines).

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Flow of liquid in narrow (capillary) channels (slots). Flow of liquid in capillary channels is of practical interest in connection with the solution of problems in the packing/seals of hydroaggregates.

In view of the fact that the clearance is measured in micrometers, flow conditions of liquid in it predominantly laminar;

therefore the calculations of friction and expenditure/consumption usually produce themselves on the basis of the condition of the laminarity of flow.

The action of similar slit gaskets is based on physical properties of liquids to resist to the flow whose value is determined that which was given [see formula (4)] by the dependence of the Newton according to which shearing stress between two layers of laminar flow proportional to the velocity gradient of velocity along standard to the axle/axis of flow.

In Fig. 20a and b is presented the diagram of flow of liquid under the action of pressure differential $\Delta p = p_1 - p_2$ between two motionless parallel plates, which are located one from another on this distance, that they form capillary slot with a size/dimension of s. Assuming that the distribution of the speeds in the section between plates has the parabolic character, which corresponds to laminar flow, and applying the relationships examined above, we find expressions for a jump/drop in the pressure Δp and in expenditure/consumption Q through similar slot [6]:

$$\Delta p = Q\frac{12\nu\rho L}{ws^3}; \quad Q = \frac{\Delta p s^3 w}{12\nu\rho L}, \qquad (22)$$

where w is width of slot in the direction, perpendicular to flow (see Fig. 20a).

After replacing in expression (22) value of $w = \pi d$, where $d = d_1 + d_2/2$ is the mean diameter of slot, we will obtain expressions for the calculation of a jump/drop in the pressure and leakages through the concentric annulus with parallel walls (Fig. 20c) during the laminar flow:

$$\Delta p = Q\frac{12\nu\rho L}{\pi d s^3}; \quad Q = \frac{\pi d\, \Delta p s^3}{12 L\nu\rho}, \qquad (23)$$

where $s = d_1 - d_2/2$ the width of slot (nominal gap length); here d_1 and D_2 are diameters of cylinder and plunger.

For the expression of the law of flow of liquid in dimensionless form, let us introduce the concept of a hydraulic radius for the annulus hearth by which we will understand the relation of the doubled cross-sectional area of flow to wetted perimeter. For annuli a hydraulic radius

$$r=\frac{2f}{\pi\,(d_1+d_2)}=\frac{2\pi}{4\pi}\cdot\frac{(d_1^2-d_2^2)}{(d_1+d_2)}=\frac{d_1-d_2}{2}=s,$$

i.e. will be equal to the nominal value of clearance s.

Expression for Re with circular concentrical slot can be presented in the form

$$\mathrm{Re}=\frac{ru}{\nu}=\frac{su}{\nu}.$$

Equating expressions (22) and (23), and also taking into account that $Q = fu = \pi dsu$ and $f = \pi ds$, we will obtain drag coefficient for a circular (concentrical) slot in dimensionless form

$$\lambda = \frac{24\nu}{su} = \frac{24}{Re}.$$

In real aggregate/units the plunger occupies relative to cylinder eccentric position, in view of which clearance a in

circumference (Fig. 20d) between them it will be variable.

Fluid flow rate through eccentric radial clearance is computed according to expression [5]

$$Q_э = \frac{\pi d \,\Delta p s^3}{12 L \nu \rho}\left(1 + \frac{3}{2}\varepsilon^2\right) = Q\left(1 + \frac{3}{2}\varepsilon^2\right),$$

where $s = d_1 - d_2/2$ - the nominal clearance with concentrical slot; $\varepsilon = \frac{e}{s}$ are a relative eccentricity; here e - eccentricity; Q - expenditure/consumption with the coaxial location of plunger and cylinder.

Taking into account that the maximum eccentricity e is equal to the nominal radial clearance s, fluid flow rate in this case

$$Q_э = \frac{2,5\,\Delta p\pi\, ds^3}{12L\nu\rho} = 2,5Q. \quad (24)$$

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Hence the pressure differential for the maximum eccentricity

$$\Delta p = \frac{12LQ_э\nu\rho}{2,5\pi\, ds^3}.$$

From the comparison of expressions (23) and (24) it is evident

that the fluid flow rate with maximum eccentricity of plunger and bushing exceeds 2.5 times expenditure/consumption in their concentrical position.

The given calculations are produced on the assumption that the viscosity of the aaaa of liquid in slot constant, whereas in actuality it depends on temperature and partially on the pressure of liquid, that are the values, alternating/variable in the course of flow of liquid. In view of the fact that a change in the temperature and, consequently, also a change of the viscosity of liquid in slot bears the complex character, which difficult yields to account during practical calculations, into the given expressions approximately introduce the average value of viscosity

$$\nu_{cp} = \frac{\nu_1 + \nu_2}{2},$$

where the ν_1 and ν_2 - the viscosity of oil at actual inlet temperatures into slot and at output/yield from it.

For flow of liquid in narrow (capillary) slots, have an effect the boundary conditions, determined in essence by the forces of molecular interaction. Under the effect of the forces, which act on the interface of liquid and of solid phases, on the walls of slot occurs the adsorption of the polar-active molecules of liquid withformation/education on them through certain time of the fixed boundary layers, which have the anomalcus viscosity, which differs in value and properties from second viscosity. Specifically, the liquid, forming this layer, acquires elastic strength tc shift/shear.

As a result of the formation/education of this fixed layer, the hydraulic slip through the slot will as a result of a reduction in area of its living section decrease in the course of time of the stay of slot under the pressure differential. With some small size/dimensions of slot, the leakage after known time can completely cease itself. As a result can be reached the complete sealing/pressurization even of the such slots whose size/dimensions hundred times exceed the molecular dimensions of working fluid and their chain/networks.

The phenomenon of the cicatrization of capillary channels in question by the layers of the adsorbed polar- active molecules,

obtain name of the obliteration of slots, is accompanied also by deposit on solid canal surface of resins, loose accumulations of the fractions of liquid, in particular the resinoid substances of colloidal character and the related with them suspensions of the solid particles of the contaminator.

Virtually the thickness of the adsorptive layers of oil on solid surface (metal), which possess high elasticity of form, capable to reliably resist the extrusion of oil from clearance, is from one to several tenths of micrometer. The thickness of the layer, which causes the obliteration of slot, taking into account the contaminations of oil, is equal for widespread brands 4-5 μm, in accordance with which complete obliteration can be observed in slots on the order of 8-10 μm.

Figure 21 gives the curves of the leakages through slots with a thickness 13 and 9 μm in the function of retention time under pressure 25 kgf/cm^2 in the oil of AMG-10.

During displacement from place of one of the surfaces, forming slot (plunger or bushings), obliterated layer is destroyed and

hydraulic slip is reduced virtually in the initial volume, whereupon process is repeated.

Hydrostatic step bearing. In a series of hydroaggregates, will find use the hydrostatic supports (step bearings) of the slips in which the lubricant was effective with as small as desired speeds, which is substantially important, for example for a hydraulic motor in the period of its launching/starting.

Fig. 21. Dependence of the leakages (expenditure/consumption) of the liquid through the capillary slot on time of experiment.

Key: (1). cm^3/s. (2). Leakages. (3). Min. (4). Time from the beginning of experiment.

Fig. 22. The design diagrams of hydrostatic heel.

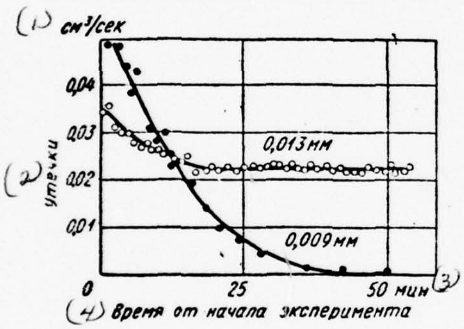

Fig. 21.

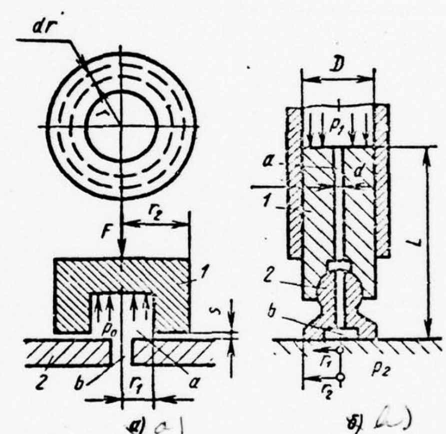

Fig. 22.

The lubricant in these step bearings is provided by feed into the capillary clearance between the flat/plane friction surfaces of the bearing of liquid under such a pressure with which external load is balanced by force of pressure of liquid and one of the sliding surfaces floats, losing in this case contact with another.

Figure 22a shows the design diagram of circular hydrostatic step bearing. For the discharging of working cell/element 1 into his chamber a will be fed through boring b in motionless bearer 2 liquid under pressure p_0 whose force it creates in the clearance between flat/plane parts 1 and 2 liquid layer, receiving the external load F. The calculation of similar supports of slip is reduced in essence to the determination of their supporting power and necessary the flow rate of the lubricating liquid.

For the calculation of the acting forces, let us use for infinitesimal cell/element dr bearing surface on radius r equation (22) for flow of liquid between parallel plates (we take the viscosity of the liquid of constant and disregard centrifugal force). After substituting in it value of $w = 2\pi r$ and $L = dr$, we will obtain

$$dp = \frac{6Q\nu\rho}{\pi r s^3} dr,$$

where Q - fluid flow rate; s is a gap length.

Integrating this equation from r_2 to r, we find pressure p in clearance on radius r (with $r = r_2$ p = 0):

$$p = \frac{6Q\nu\rho}{\pi s^3} \int_r^{r_2} \frac{dr}{r} = \frac{6Q\nu\rho}{\pi s^3} \ln \frac{r_2}{r}. \tag{25}$$

With $r = r_1$ $p = p_0$, as a result of equation for the calculation of pressure p_0 and of flow rate Q, the liquids through the flat/plane slot in question take the form

$$p_0=\frac{6Q\nu\rho}{\pi s^3}\ln\frac{r_2}{r_1};\; Q=\frac{\pi p_0 s^3}{6\nu\rho\ln\frac{r_2}{r_1}}. \qquad (26)$$

After dividing expression (25) for expression (26), we will obtain

$$p=\frac{p_0\ln\frac{r_2}{r}}{\ln\frac{r_2}{r_1}}.$$

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The load-lifting force F of step bearing will be determined by the sum of forces of pressure p_0 of chamber a ($p_0 \pi r_1{}^2$) and variable according to radius r clearance pressure s on the mating surfaces:

$$F = p_0 \pi r_1^2 + \int_{r_1}^{r_2} p \cdot 2\pi r dr \ .$$

After substituting p from expression (25) into the last/latter expression, we will obtain

$$F = p_0 \pi r_1^2 + \frac{12 \mu Q \nu \rho}{s^3} \int_{r_1}^{r_2} \ln \left(\frac{r_2}{r_1} \right) r dr \ .$$

After substituting from equation (26) Q and after integrating,

we will obtain expression for the calculation of the load-lifting force of the circular hydrostatic step bearing

$$F=p_0\frac{\pi}{2}\cdot\frac{r_2^2-r_1^2}{\ln\frac{r_2}{r_1}}.$$

In the majority of the constructions of pumps and hydraulic motors, hydrostatic supporting/reference step bearings (shoes) are supplied from their working medium without the application/use of an auxiliary source of pressure. one of similar supports, used in axial rotor- piston pumps in the places of the coupling of the piston crown and inclined washer, is represented in Fig. 22b. The supply of liquid from working pump-stock to the sliding spherical and flat surfaces is realized through the choke axial channels a of the piston of 1 and seating shoe 2.

This shoe receives the force of operating pressure p_1 liquid in cylinder on piston 1. The equilibrium condition of acting body forces

takes the form

$$p_1 \frac{D^2}{2} = p_2 \frac{r_2^2 - r_1^2}{\ln \frac{r_2}{r_1}},$$

where p_1 and p_2 - pressure in pump-stock and pressure in chamber b of seating shoe 2.

Fluid flow rate through the end-type slot can be calculated for the selected clearance according to equation (26) with replacement p_0 by p_2:

$$Q = \frac{\pi p_2 s^3}{6 \nu \rho \ln \frac{r_2}{r_1}}.$$

By equating this flow rate on the flow rate through the axial

channel a in piston, computed from equation (13), we will obtain equation for determining diameter d of opening/aperture

$$\frac{(p_1 - p_2)\, d^4}{128L} = \frac{p_2 s^3}{6 \ln r_2/r_1};$$

$$d = \sqrt[4]{\frac{21{,}3 p_2 s^3 L}{(p_1 - p_2) \ln r_2/r_1}}.$$

Hydraulic impact.

In connection with the application/use of high rates of flow of liquids in the conduit/manifolds of hydraulic systems (in a number of cases these rates reach 30 m/s) and the propagation in them of the high-speed distributors (switching rates are led to thousandths of a second) important value they acquire the questions, connected with the hydraulic impact with which the pressure overshoots can in several (three and more) once to exceed the nominal operating

pressure by hydraulic system. Similar excess/throw/overshoots lower the service like of the work of hydroaggregates and conduit/manifolds, but sometimes they can cause their failure.

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Hydraulic shock in the general case they call the oscillation of the pressures, which accompany any transient process (the transient conditions of flow) in liquid from one mode/conditions to another, caused, for example, by control of hydraulic mechanism or by another change in the mode/conditions of its work.

Hydraulic impact is caused by the compressibility of liquid and by the elastic deformation of conduit/manifold, and also by the distributivity of the mass of liquid along the length of conduit/manifold.

From entire diversity of the possible disturbance/perturbations, which produce hydraulic impact, the greatest practical interest they represent the disturbance/perturbations, caused by an abrupt change

in the rate of flow and pressure of liquid.

The change (reduction) in the speed of flow, which is accompanied by hydraulic impact, is caused in essence by the overlap of the channels (main lines) of hydraulic system in the process of distribution and flow-rate control of liquid by equipment for hydraulic system. Thus, for instance, tests will show that during the changeover of distributors with positive overlap and the valve discharging of pump the excess/throw/overshoots at operating pressures 100 kgf/cm² reach 250 kgf/cm².

The calculation of an impact (excess/throw/overshoot) in the pressure increase produces, utilizing the equation of kinetic energies, according to which the kinetic energy of the driving liquid is converted into the strain energy of the walls of duct and compression of liquid. For the case of the instantaneous and complete overlap of the rectilinear simple conduit/manifold, filled by the driving liquid, an impact pressure increase can be calculated according to the known equation of N. Ye. Joukowski [12]

$$\Delta p_n = \rho u_0 a, \qquad (27)$$

where ρ and a are density of liquid and the velocity of shock wave; u_0 is the initial velocity of the motion of liquid in conduit/manifold (prior to the beginning of the overlap of conduit/manifold).

Hydraulic impact is accompanied by wave process in conduit/manifold with the imposition of shock waves, during which fluctuations (pulsation) of pressure are repeated until the initial kinetic energy is absorbed by friction (converted into heat).

The preceding/previous expression will be correct, if the overlap of conduit/manifold occurred "instantly", i.e., for the case when time t of overlap is shorter than the phase of shock τ whose hearth is understood time of landing run by the shock wave of the dual length of the section of conduit/manifold in question (from

overlapping gate to the source of flow rate and vice versa):

$$t < \tau = \frac{2L}{a}, \quad (28)$$

where L it is understood the length of the section of conduit/manifold in question.

Under this condition the overlap of conduit/manifold concludes before the reverse/inverse shock wave, reflected from the source of flow rate (pump, storage battery/accumulator, etc.), it will return to gate. Hydraulic impact for this case will be determined by the total loss by the liquid of velocity, in accordance with which the pressure overshoot will be maximum. Hydraulic impact in this case is conventionally designated as complete or straight line.

From the aforesaid it follows that the pressure overshoot with direct/straight hydraulic impact will achieve extreme value only on that section of the conduit/manifold, counting from gate, on which will have time to be extended the direct wave, which appears at the

torque/moment of the complete coverage of gate, to its rendezvous with the backward wave, reflected from the source of flow rate.

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From equation (28) it follows that maximally the possible for the emergence of direct/straight hydraulic impact time of the overlap of conduit/manifold

$$t = \frac{2L}{a}.$$

The maximum impact pressure, equal to impact pressure during the instantaneous overlap of conduit/manifold, will be observed at this value of t of gate itself. In remaining sections with approach/approximation to the source of flow rate (storage battery/accumulator) the pressure descends up to a pressure of the

latter.

Under condition

$$t > \tau = \frac{2L}{a},$$

i.e. with slower than it is given above, the overlap of conduit/manifold, an impact pressure increase will be determined only by the that part of the initial velocity of liquid $\Delta u = u_0 - u$, which will be lost for the time, equal to period τ conduit/manifold. Under this condition reverse/inverse shock wave, after being reflected from the source of flow rate, will return to gate earlier than conduit/manifold will be completely overlapped. A similar shock is conventionally designated as indirect or incomplete.

An impact pressure increase in this case will be determined by expression

$$\Delta p_{\text{н}} = \rho \Delta u a,$$

where $\Delta u = u_0 - u$ - the decrease in the velocity of liquid in duct, caused by the partial overlap of it by tap/crane for the time, equal to period τ conduit/manifold;

here u_0 is the initial velocity of the motion of liquid (velocity prior to the beginning of the overlap of conduit/manifold);

u - the changed velocity of liquid (velocity up to the torque/moment of arrival at the gate of the reverse/inverse shock wave, reflected from the source of flow rate).

After assuming that a change in the speed of flow in conduit/manifold flow/lasts evenly, the calculated speed loss Δu for time τ can be approximately calculated according to expression

$$\Delta u = \frac{u_0 \tau}{t}.$$

The excess/throw/overshoot of the pressure of $\Delta p_{н}$ with indirect (incomplete) shock ($t > \tau$) can be calculated also according to expression

$$\Delta p_{н} = \frac{\tau}{t} \Delta p_{п}.$$

Taking into account the preceding/previous equations the last/latter dependence can be represented in the form

$$\Delta p_{н} = \frac{2\rho L u_0}{t}. \tag{29}$$

Velocity of shock wave. The entering the resulting expressions velocity a of shock wave in the elastic liquid, included into conduit/manifold with elastic walls $(E \neq \infty)$, is determined from the equation of N. Ye. Joukowski [12]

$$a=\frac{1}{\sqrt{\rho\left(\frac{1}{K}+\frac{d}{Es}\right)}}=\frac{\sqrt{\frac{K}{\rho}}}{\sqrt{1+\frac{dK}{Es}}}, \qquad (30)$$

gde ρ - the density of liquid;

K - the bulk rigidity modulus of liquid (for the deaerated mineral oil $K = 1.6 \cdot 10^4$ kgf/cm²);

d and s - inner diameter and the wall thickness of conduit/manifold;

E - the modulus of elasticity of the material of duct (for a conduit/manifold made of steel of 1X18N9T $E = 2 \cdot 10^6$ kgf/cm²).

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In accordance with this, expression (27) for the calculation of impact pressure in the elastic conduit/manifold of $(E \neq \infty)$ with complete (straight line) shock will be presented in the form

$$\Delta p_n = \rho u_0 a = \frac{\rho u_0}{\sqrt{\rho\left(\frac{1}{K} + \frac{d}{Es}\right)}} = \frac{\rho u_0 \sqrt{\frac{K}{\rho}}}{\sqrt{1 + \frac{dK}{Es}}}.$$

For a conduit/manifold with the absolutely rigid walls of $(E = \infty)$ the velocity of propagation of shock wave is equal to the speed of propagation of sound in this liquid medium with density ρ and bulk rigidity modulus k:

$$a = \sqrt{\frac{K}{\rho}}.$$

This speed for the deaerated mineral oil a = 1320-1440 m/s, for the being applied in hydraulic systems oil mixture of AMG-10 with t = 20°C a = 1290 m/s.

When, in the oil, the undissolved air is present, in formula (30) substitutes itself for the bulk rigidity modulus of liquid the given bulk rigidity modulus of the mixture of liquid with air. In accordance with this case it will be below than with the deaerated liquid. This is caused by the fact that when, in the liquid, the undissolved air is present, the latter, being compressed during a pressure increase, allow/assumes certain displacement of liquid and thereby reduces period τ (phase of shock) conduit/manifold.

Hydraulic impact in diversion/taps. A pressureincrease with hydraulic impact in any branch of hydraulic system produces hydraulic impact in all diversion/taps from it and in particular into dead-end ones.

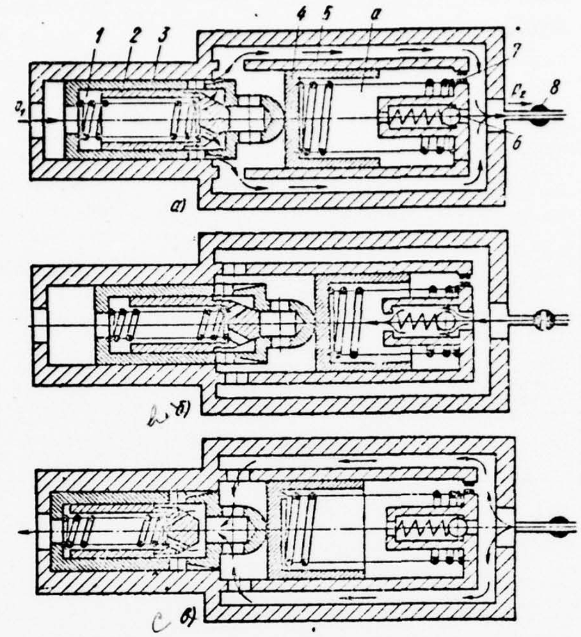
This is caused by the fact that the pressure, which arose in conduit/manifold with hydraulic impact, being spread on diversion/tap, contributes as a result of the deformation of its walls and liquid to the motion of the latter to blind alley. As a result are created analogous conditions for developing wave process, as in the case of the overlap of conduit/manifold with the driving liquid.

Analogous hydraulic impact is observed also with the instantaneous (abrupt) connections of the blind main lines or other rigid tanks, filled by liquid, to the source of higher pressure (to the working main line of hydraulic system, pneudraulic storage battery/accumulator, etc.).

In this case, if time t of discovery/opening tap/crane with the connection of dead-end conduit/manifold is shorter than the period τ conduit/manifold, then the pressure, which develops as a result of hydraulic impact in blind alley, exceeds the perturbation pressure (pressure before the tap/crane) virtually 2 times.

Hydraulic impact in blind diversion/tap has great practical value, since any of the inoperative at any torque/moment main lines of the branched delivery conduit can be considered as the blind diversion/tap, gate in which is created by the connected at its end any aggregate/unit, which overlaps this diversion/tap. Such blind diversion/taps include also the main lines of the connection of the different measuring and monitoring instruments: manometers, indicators,etc.

Fig. 23. Diagrams of the extinguisher of hydraulic impact.



Methods of a reduction in the impact pressure. To soften hydraulic impact possible either by an increase in time t of the changeover of distributor (overlap of conduit/manifold) to

$$t > \frac{2L}{a},$$

or by the decrease τ, realized usually with the aid of the different compensators (extinguishers) of shock.

The timing of the changeover of distributor usually is realized by the battery supply relays, during application/use of which it is possible to ensure the required for unstressed changeover time t.

Diagram of one of a similar relay, intended for smooth pressure balance during the instantaneous connection of two main lines with the aid of the high-speed pet cock (gate) 8, established/installed on output/yield from relay, is given in Fig. 23.

Relay consists of choke plunger valve 3 in which is placed check valve 2, and dosage piston 4, stretched spring 5.

The position of the moving elements of the relay, presented in Fig. 23a, corresponds to discovery/opening pet cock 8. Plunger valve 3 under the action of the created pressure differential $\Delta p = p_1 - p_2$ in input and outlet ducts is moved to the right, open/disclosing the passage slot, formed with the conical part of valve 3 and by the housing of relay, through which the liquid proceeds to outlet duct and further tap/crane 8.

In the initial stage of the displacement of valve 3, section of passage slot and, consequently, also fluid flow rate through it are close to zero, and only after passage by it certain assigned way the section of slot and fluid flow rate smoothly reach the required (calculated) value.

In the final (extremely right) position of valve 3, the friction of relay to fluid flow is determined only by the force of return spring 5.

The speed of valve opening 3 and respectively the rate of an increase in the flow rate through the forming in this case expenditure slot is determined, other conditions being equal, by the speed of the displacement of piston 4 of measuring apparatus, assigned by the friction of throttle/choke 7 through which is displaced the liquid from chamber a in moving piston 4.

The position, presented in Fig. 23b, corresponds to the overlap of conduit/manifold by tap/crane. Flow of liquid in the input and outlet ducts of relay ceases, and its movable parts (valve 3 and piston 4) under the action of return spring 5, they are moved at starting positions. For the acceleration of this process, is provided for check (locking) valve 6 through which the liquid overflows in parallel with the flow rate through throttle/choke 7 into the right cavity of chamber a of piston 4, providing its quick traverse at starting position (to the left).

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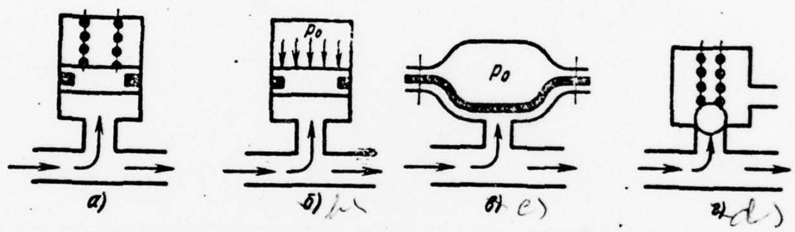

Fig. 24. Diagrams of the compensators of hydraulic impact.

During a change in the direction of fluid flow (Fig. 23c) relay works as the support valve whose friction (backwater) is determined by the force of spring 1, which loads the lock of valve 2.

It is obvious, by means of the selection of the corresponding friction of throttle/choke 7 and of the angle of the conical part of valve 3 it is possible to ensure the required time of the connection of input and exit main lines, and consequently, to ensure smooth pressure balance in these main lines.

The required time t of the coverage (or the discovery/opening) of gate during which will be provided for the assigned excess/throw/overshoot of the pressure of Δp_{κ}, it can be determined from expression (29).

Practice shows that freedom from impact/shock of the connection of main lines with the pressure differential 220 kgf/cm² reliably provided with $t \geqslant 0.1$ s.

Compensators of hydraulic impact. The compensator (extinguisher)

of hydraulic impact usually is the connected with conduit/manifold container of one form or the other (Fig. 24) with elastic cell/element. Reduction by the compensator of impact pressure occurs as a result of absorption with the deformation of its elastic cell/element of certain part of the energy of the shock wave, which enters the compensator in the form of the fluid flow, which corresponds to a velocity increment in the shock wave above the initial velocity.

Are common piston compensators with spring (Fig. 24a) and gas (Fig. 24b) elastic cell/elements.

The pressure of charge by gas of gas compensator (Fig. 24b) usually is selected equal (or somewhat above) to the maximum operating pressure in hydraulic system.

Compensator to working main line hydraulic systems connect by the tube of the smallest possible length and maximum section, which is dictated by effect on the dynamic process of the connected (given) mass of liquid.

This effect is caused by the fact that the inertia pressure p, which appears during the unsteady motion of liquid, is raised proportional to an increase in length L of the acceleration/dispersal of the elementary particle of the liquid:

$$p=\rho L\frac{du}{dt},$$

where du/dt is proportional the particle acceleration of liquid.

Timing of the function (operating speed) of a similar system shows that the effect of the reduced mass of liquid in many instances (with long connecting conduit/manifold and its small diameter) it is considerable (5-6 times) it predominates above the effect of the mass of the movable mechanical features of the compensator (piston, etc.).

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An increase 2 times of the diameter of conduit/manifold is accompanied by decrease 4 times of the reduced mass of liquid.

The reduced mass of liquid in connection with cylindrical compensator

$$M_{np} = m\left(\frac{D}{d}\right)^4,$$

where m is a mass of liquid; D and d is a diameter of the cylinder of compensator and connecting conduit/manifold.

A deficiency/lack in the piston compensators is the large inertness, caused by the mass of piston, and also the presence of friction in cylinder. As a result of this, and also due to the inertness of elastic liquid column in the conduit/manifold, which connects the liquid reservoir of compensator with the working main line of hydraulic system and with gutter, the piston of compensator

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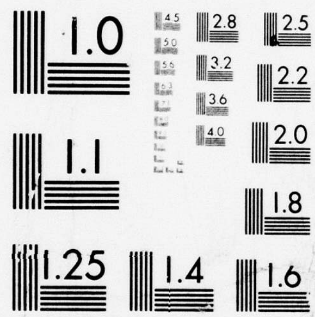

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can initiate (as a result of the blast effect or pulsation of pressure in system) to oscillate, which can lead to an asynchronous with respect to blast effect sign change the direction of the motion of liquid in this channel (to the appearance of a "negative" velocity). In this case, the pressure in connecting conduit/manifold can exceed blast pressures in the shielded main line, as a result of which a similar compensator not only will not absorb wave energy, but aggravate its action, which will lead to an increase in the pressure overshoots.

For a decrease in the inertness of the movable cell/element of compensator liquid and gaseous medium they divide by elastic rubber blanket (Fig. 24c). Capacity is selected from the conditions of the equality of the compression work of gas in compensator and the kinetic energy of liquid, which takes place on conduit/manifold. Virtually the capacity of gas cavity usually is 200-250 cm^3.

For the limitation of impact pressure, they apply also protective (Fig. 24d). For an increase in the effectiveness of valve, it is necessary to decrease the inertness of the movable cell/elements of valve and liquid which depends on the mass of the moving elements of the valve, and also of section and length of the

conduit/manifolds with the aid of which the valve is connected to working main line, and also the drainage conduit/manifold, which connects valve with tank.

It is obvious, most effective under otherwise equal conditions it will be the setting up of valve directly on working main line with the jettisoning of liquid into drainage large-slot manifold.

For the extinguishing of hydraulic impact, are suitable only the action of direct valves (Fig. 88), but not two-stage valves (valves with servo effect), which differ from the first in terms of more prolonged pause (delay) between the feed of the signal of pressure and discovery/opening the basic (discarding) valve, caused by the double graduation of action and by the inertness of liquid on the channels of small sections.

Hydrodynamic pressure of liquid jet on wall. Practical interest, in particular during calculations of distributors of the type nozzle - shutter/valve, represents the force of pressure of fluid flow, which escape/ensues of nozzle to the barrier/obstacle (wall), arrange/located in magnetic path (Fig. 25).

The reaction of liquid jet to wall in the assigned direction is measured by projection on this direction of a change in the momentum. The force of the effect of flow to flat/plane motionless wall more than six diameters in diameter of jet cross-sectional area, arrange/located perpendicular to flow direction, will be equal on the basis of the theorem of mechanics about momentum to the power impulse per second:

$$P = mu = Qu\rho,$$

where m and Q it will be the mass and volumetric fluid flow rate; u is the average speed of fluid flow.

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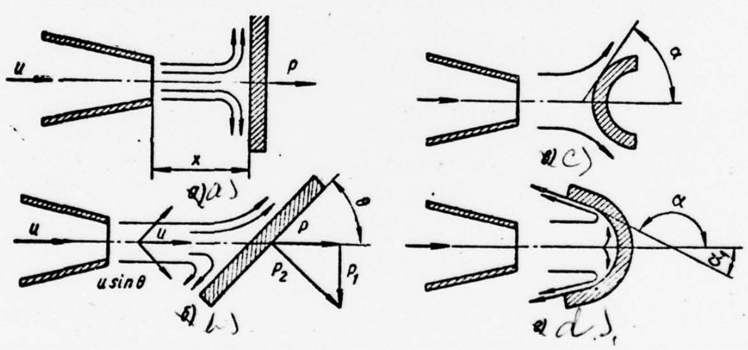

Fig. 25. Diagram of the action of fluid flow to riding-crops (shutter/valves).

Taking into consideration that

$$Q = \omega u = \frac{m}{\rho},$$

we can write

$$P = \rho u Q = \rho \omega u^2,$$

where ρ - the density of liquid; ω is flow cross section.

Actual force depends on distance x between the nozzle edge and the wall, decreasing during the removal/distance of nozzle from barrier/obstacle. The latter is caused by scattering the energy of jet.

During the fixed array of wall at an angle θ for flow direction

(see Fig. 25b) this force

$$P_2 = Q\rho u \sin\theta;\ P = Q\rho u \sin^2\theta;\ P_1 = Q\rho u \sin\theta\cos\theta.$$

If wall is moved in the same direction, as jet, at a rate of v, then the speed of the rendezvous of jet with wall decreases in relation

$$\frac{u-v}{u}.$$

In accordance with this for the first case of the location of wall (see Fig. 25a)

$$P = \frac{\rho Q}{u}(u - v)^2 = \rho\omega(u - v)^2.$$

Speed v of the displacement of wall usually disregard in view of its relative smallness.

For the case of the effect of jet on the curvilinear plate (hemisphere) of comparatively small size/dimensions (we take the speed of flow equal to speed of the nozzle outlet) the force (Fig. 24c)

$$P = \rho Q u\,(1 - \cos\alpha).$$

At the obtuse angle α (Fig. 24d), equal to $180 - \alpha_1$,

$$P = \rho Q u \,(1 + \cos \alpha_1).$$

With an increase in the angle α, the pressure of jet on wall grow/rises, reaching during the complete rotation of jet ($\alpha = 180°$ and $\alpha_1 = 0$)

$$P = 2\rho Q u.$$

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CHAPTER II. Actuating mechanisms.

As actuating mechanisms (hydraulic engines) are applied the actuating cylinders, employed for the realization of reciprocating rectilinear and rotary displacement/movements, and also the hydraulic motors of the continuous rotary motion, which convert the energy of fluid flow respectively into the rectilinear forward/progressive,

rotary rotary displacement/movement of exit stock/rod (shaft).

MECHANISMS OF RECTILINEAR MOTION.

As the actuating mechanisms of rectilinear motion, are applied predominantly the actuating cylinders (see Fig. 3a).

Figure 26 gives the diagrams of the cylinders of two basic types: bilateral (a and b) and one-sided (c) action; the piston (plunger) of the last/latter cylinder accomplishes back stroke under spring effect or external forces.

The driving/moving force P on the stock/rod of cylinder and the speed v of his displacement/movement not allowing for losses for friction, counterpressure and hydraulic slipes determine from formulas

$$P = pf \text{ и } v = \frac{Q}{f}, \quad (31)$$

[И = and]

where f is the effective area of piston; Q - volumetric fluid flow rate; $f = \pi D^2/4$ - the effective area of piston for the cylinder, presented in Fig. 26a during the supply of liquid into the cavity, opposite to stock/rod; $f = \pi (D^2 - d^2)/4$ - the effective area of piston for the cylinder, presented in Fig. 26a, during the supply of liquid into cavity stock/rod's on the part and for the cylinder, presented in Fig. 26b; here D and d are diameters of piston and stock/rod.

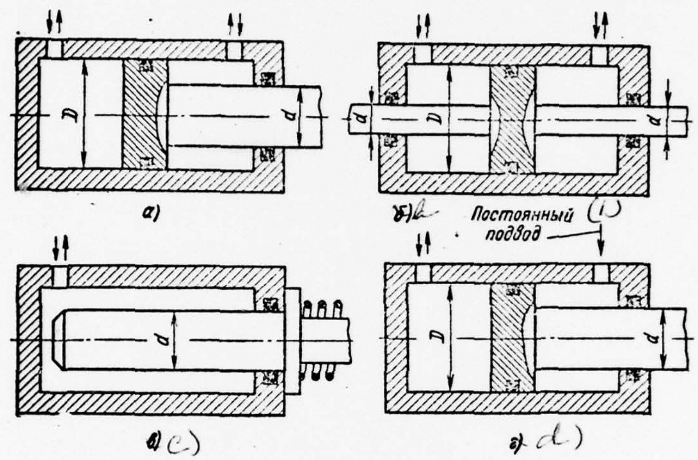
For the cylinder of one-sided action (Fig. 26c) effective area is the sectional area of stock/rod (plunger)

$$f = \frac{\pi d^2}{4}.$$

These cylinders are simple in production, since to treatment/working is subject only surface d of bush under stock/rod (plunger) and there is no need for for working the surface of inner cylinder face.

Fig. 26. Types of actuating cylinders.

Key: (1). Constant supply.



If cylinder with one-sided stock/rod is include/connected into hydraulic system then so that his rod cavity would be constantly connected to main pressure line, and opposite cavity alternately either to this main line or to drainage (Fig. 26d), then it will seem possible to carry out both equal speeds with direct/straight and back strokes and different.

During the connection/compound of left cavity with tank, the piston under the action of the constant forces of the pressure of liquid on the right cavity will be moved to the left, during its connection/compound with main pressure line the piston as a result of a difference in the effective areas will be moved to the right.

Figure 26d shows that with

$$d = \frac{D}{\sqrt{2}}$$

stock/rod's area will be 2 times less than the area of cylinder. Effective areas during piston stroke into both sides in this case

will be equal and determined from expression

$$f = \frac{\pi D^2}{4} - \frac{\pi (D^2 - d^2)}{4} = \frac{\pi d^2}{4}.$$

In accordance with this piston speed and developed with it force with course into both sides will be also equal.

With $d \neq D/\sqrt{2}$ the effective area is equal: $f = \pi (D^2 - d^2)/4$ - during piston stroke to the left; $f = \pi d^2/4$ - during piston stroke to the right.

With $d > D/\sqrt{2}$ we will obtain high speeds of back stroke (to the left) and the large forces of direct/straight power stroke (to the right).

Figure 27 depicts the analogous diagram of constant power supply of one of the cavities of cylinder by means of the connection to it of hydropneumatic storage battery/accumulator. Pump in this diagram connect through distributor 1 with the rod (left) cavity of actuating cylinder 2, and storage battery/accumulator 3 connect with opposite (right) cavity. During supply through the distributor of 1 liquid from pump into the left cavity of cylinder 2, which corresponds to his working course, piston is moved and displaces to the right liquid from right cavity of storage battery/accumulator 3.

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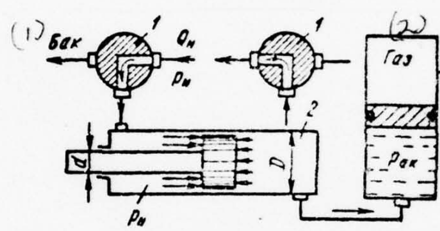

Fig. 27. Diagram of the power supply of actuating cylinder by hydropneumatic storage battery/accumulator.

Key: (1). Tank. (2). Gas.

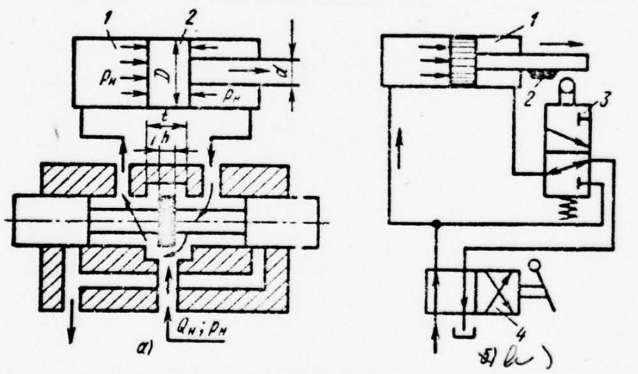

Fig. 28. The actuating cylinder, equipped with distributor with negative overlap (a), and the circuit diagram of this cylinder into

hydraulic system (b).

During the connection/compound of the left cavity of cylinder with tank (in the position of distributor, shown to the right) piston under the pressure of the liquid, which enters from storage battery/accumulator 3 the right cavity of cylinder 2, is moved to the left (back stroke of piston).

With the power supply of the left cavity of cylinder by liquid under the pressure of $p_н$ the force, developed with piston,

$$P=\frac{\pi(D^2-d^2)}{4}p_н-\frac{\pi D^2}{4}p_{ак}=\frac{\pi}{4}(D^2-d^2)p_н-D^2p_{ак}, \qquad (32)$$

where the $p_н$ - the pressure of power supply (forcing): $p_{ак}$ - pressure in storage battery/accumulator; D and d are diameters of cylinder and stock/rod.

With the power supply of cylinder by storage battery/accumulator force (pressure of gutter we disregard)

$$P=\frac{\pi D^2}{4}p_{ак}. \qquad (33)$$

Piston speed with foreward stroke (with the power supply by pump) will be determined

$$v_n = \frac{4Q_H}{\pi (D^2 - d^2)}, \qquad (34)$$

where the Q_H - the feed of pump.

Since the energy of storage battery/accumulator can be used during short time, the back stroke of piston can be completed in so short a time as this admissibly for the present instance of applying

a hydraulic system.

Since the pressure of $p_{ак}$ in storage battery/accumulator during piston stroke as a result of a change in the gas volume with the displacement of liquid from the right cavity of cylinder (with piston stroke to the right) and power supply (with piston stroke to the left) it will be variable, during the calculation of p_{a}, one should accept its maximum value of $p_{ак\,max}$ with the direct/straight (pump) piston stroke of cylinder 2 according to expression (32) and the minimum $p_{ак\,min}$ - with back stroke according to expression (33).

The quick traverses of the piston of actuating cylinder with one-sided stock/rod can be ensured by the connection/compound of both cavities of cylinder with one-sided stock/rod. Figure 28a depicts one of the diagrams of this system, equipped with distribution valve with negative overlap (h < t) in the mid-position (see also Fig. 44).

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In the end (left and by right) positions of the plunger of valve, liquid entering from pump, is directed to respectively for the right or left cavity of actuating cylinder 1, providing the speed of the displacement/movement of its piston 2, which corresponds to the working section of these cavities [see expression (31)]. In the average/mean position of the plunger, the channels, which drive into tank, are blocked (they overlap), and both cavities of cylinder 1 are connected between themselves and with the channel of the pump through the slot of the negative overlap of valve, in view of which the pressures of liquid in them will be equal (hydraulic friction we disregard). As a result piston 2 under the action of that not balanced as a result of a difference in the working area the action on it of the pressure of liquid is moved to the right.

Since the liquid, displaced in this case of the right cavity of cylinder, enters its left cavity, the piston speed of v_n will be determined by the feed of the $Q_н$ of pump and by the feed, displaced from the right cavity of the cylinder of $Q_{цил}$.

Force, developed on piston,

$$P = \frac{\pi D^2}{4} p_н - \frac{\pi (D^2 - d^2)}{4} p_н' = \frac{\pi d^2}{4} p_н.$$

In accordance with this working (uncompensated for) piston clearance in this diagram, is the sectional area of stock/rod

$$f = \frac{P}{p_{н}} = \frac{\pi d^2}{4}.$$

Consequently, the piston speed in this case

$$v_n = \frac{Q_{н}}{f} = \frac{4}{\pi} \cdot \frac{Q_{н}}{d^2}.$$

In the systems of automation, the indicated connection/compound of the cavities of actuating cylinder on the required part of the course of its piston usually is realized with the aid of the supplementary three-way distributor (switch), controlled by any external unit (detent or electric switch). Figure 28b shows the diagram of system with similar supplementary distributor 3 and cam/catch/jaw 2, established/installed on the stock/rod of cylinder 1. The left cavity of cylinder is connected through basic distributor 4 with forcing, and right through the supplementary with gutter, as a result the piston is moved to the right at a velocity, which corresponds to its working area from pressure side. However, after moving himself stock/rod will embed with the aid of the established/installed on it cam/catch/jaw the plunger of distributor 3, the left and right cavity of cylinder they are connected between

themselves and with forcing, as a result the piston will be moved in the same direction, but at a velocity, determined by its stock/rod's section. On the break-down of detent on the plunger of distributor, it again will move with the aid of spring at the initial overhead position (depicted on Fig. 28b), in this case the cavity of cylinders again they will be separate/liberated from each other.

The examined connection/compound is applied in the systems in which it is required to ensure, for example, the in-rapid traverse of cutting tool to workpiece. By adjustment on the stock/rod of the cylinder of several detents is represented possible to obtain the alternation of the retarded and quick traverses of the stock/rod the need for whom appears, for example, with the bore of multiweb parts.

Are applied also the power cylinders of compound circuits (with stepped piston, telescopic cylinders, etc.).

Figure 29a depicts the diagram of cylinder with the stepped pistor with the aid of which it is possible to obtain several speeds.

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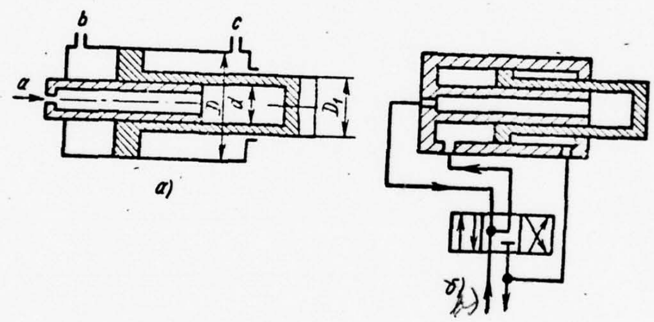


Fig. 29. Actuating cylinder with stepped piston (a) and the diagram of the power supply of this cylinder (b).

During the supplying of the liquid of $Q_н$ with the pressure of $p_н$ into channel a, we will obtain maximum speed

$$v_1 = \frac{4Q_н}{\pi d^2}$$

and the minimum force

$$P_1 = \frac{p_н \pi d^2}{4},$$

during the supply of liquid into channel b we will obtain the average speed

$$v_2 = \frac{4Q_н}{\pi (D^2 - d^2)}$$

and the force

$$P_2 = \frac{\pi (D^2 - d^2)}{4} p_н.$$

With simultaneous feed into channels a and b, we will obtain minimum speed

$$v_3 = \frac{4Q_н}{\pi D^2}$$

and the maximum force

$$P_3 = \frac{\pi D^2}{4} p_н.$$

Speed of back stroke (during the supplying of liquid into channel c) and the force:

$$v_4 = \frac{4Q_H}{\pi (D^2 - D_1^2)}; \quad P_4 = \frac{\pi (D^2 - D_1^2)}{4} p_H.$$

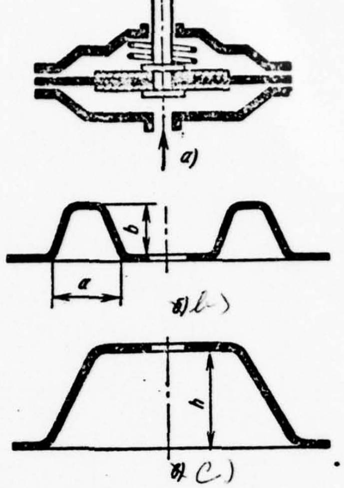
The diagram of the power supply of this cylinder is shown in Fig. 29b.

Mechanisms with flexible separators.

For the realization of small rectilinear displacement/movements in small forces, are applied actuating mechanisms (hydraulic engines) with elastic separators in the form of flat/plane (Fig. 30a) or figure (Fig. 30b and c) rubber-fabric diaphragm/membranes. With the aid of these diaphragm/membranes it is possible to ensure the complete airtightness of connection/compound and simultaneously small friction, thanks to which membrane/diaphragm mechanisms they found use both in hydro- and in pneumatic systems at small (5-10 kgf/cm^2) pressures.

Flat diameter (see Fig. 30a) it differs in terms of simplicity; however, its effective area changes during the displacement/movements of center more intense than the effective area of diaphragm/membranes with figure elastic part (see Fig. 30b and c). Furthermore, the flat diameter allow/assumes the considerably less course of center in comparison with figure, which allow/assume certain sagging/deflection of diaphragm/membrane without the elongation of fabric.

Fig. 30. Diagrams: a) membrane/diaphragm type actuating mechanism: b and c) diaphragm/membranes.



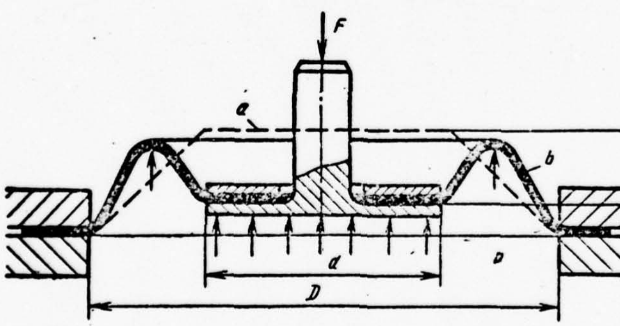


Fig. 31. Diagram of actuating mechanism with rubber-fabric diaphragm/membrane.

Figure 31 depicts the diagram of drive with the dished diameter, initial form of which is shown by broken lines. Under the action of load F and of forces of pressure, the diaphragm/membrane accepted the form, shown in the diagram.

The maximum course of dished diameters (see Fig. 30c) is approximately equal to their dual height/altitude h. The maximum course of the corrugated diaphragms of diaphragm/membranes (see Fig. 30b) is equal approximately to width a ridge or to its dual height/altitude b. The course of flat diameters (see Fig. 30a) must not exceed 7-10o/o of diameter D of the circumference of their stopping up.

Membrane/diaphragm hydraulic (pneumatic) actuating mechanism is the pinched on periphery elastic ring b, with center of which is connected the load (see Fig. 31). As a rule, this ring has the rigid center, diameter d of which is 0.75-0.85 diameters D of the jamming of ring in housing.

In automatic hydraulic-pneumatic equipment are common also the metallic corrugated diaphragms (from beryllium bronze or

chrome-nickel alloy) symmetrical (Fig. 32b) types. Symmetrical diaphragm/membrane is obtained by means of the welding of two asymmetric diaphragm/membranes.

The strain of such diaphragm/membranes is the function of a jump/drop in the pressure $\Delta p = p_1 - p_2$ and in the external load F. Strain comprises to 5o/o for the diaphragm/membranes symmetrical and 2-3o/o of the diameter of stopping up for the diaphragm/membranes asymmetric types.

Figure 32c gives the curves of dependences $x = f\ (p)$ for symmetrical and Fig. 32d - for asymmetric the types of the diaphragm/membranes which show that the characteristics have certain virtually linear section AB.

Force in the center of diaphragm/membrane. The important parameter of membrane/diaphragm device is the effective area of the diaphragm/membrane which determines the developed in the center of diaphragm/membrane force in the direction, perpendicular to the plane of its jamming. Under the effective area of diaphragm/membrane in the general case, is understood this area, which while that which was

multiplied by the pressure differential, which acts on diaphragm/membrane, determines the force, developed in its center. It is obvious, this determination it does not reveal the physical sense of effective area, since on diaphragm/membrane itself there is no figure, geometric area of which would determine its effective area.

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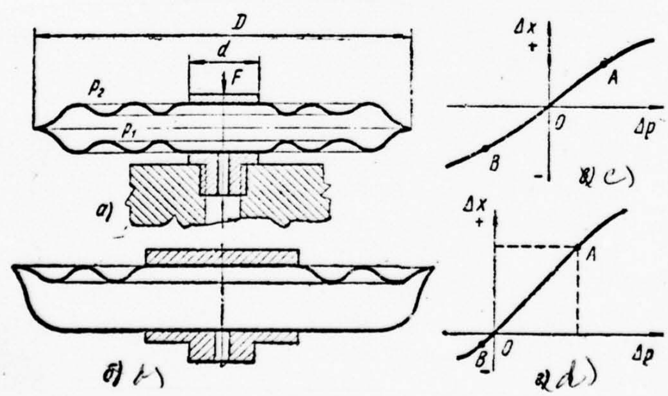
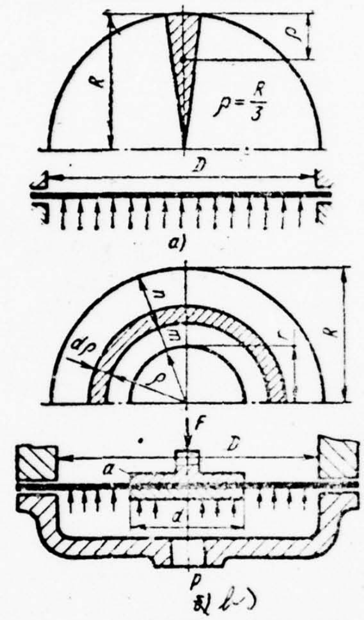


Fig. 32. Metallic are diaphragm/membranes (a and b) and their characteristics (c and d).

Fig. 33. The design diagrams of flat diameter.



Effective area depends both on the design parameters and on a series of other factors (course of the center of diaphragm/membrane and the hardness its material, a jump/drop in the pressures, etc.), the determination of effect of which is usually difficult. In view of this effective area they calculate with small (close to zero) sagging/deflections according to the approximation formulas with the subsequent experimental check.

Simplest is the actuating mechanism in which the diaphragm/membrane on has rigid center. The design diagram of this mechanism is shown in Fig. 33a. Surface area S of diaphragm/membrane can be conditionally broken into a series of the elementary isosceles triangles with apex/vertexes in the center of diaphragm/membrane and basis/bases at the duct of their attachment. In this case, it is possible to assume that resultant force of the pressure of working medium on each triangle is applied at a distance ρ, equal to 1/3 height/altitudes R of triangle from basis/base. The respectively bearing pressure of elementary triangle with an area of S_0 will be inversely proportional to distance from support to the point of the application/appendix of resultant. It is obvious that the sum of bearing pressure with this assumption will be approximately equal to 2/3pS of the duct of jamming and 1/3pS of the center of diaphragm/membrane.

In accordance with this force F of the pressure of working medium, transferred to the center of this diaphragm/membrane, fastened on outer duct, with its location in plane (with zero sagging/deflection)

$$F = \frac{1}{3} pS,$$

where p is the pressure differential, which act on diaphragm/membrane; $S = \pi D^2/4$ - the total area of diaphragm/membrane according to diameter D of the attachment of duct in housing.

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In accordance with this the effective area of the $S_{эф}$ of the diaphragm/membrane, which does not have rigid center,

$$S_{э\phi} = \frac{F}{p} = \frac{1}{3} S = \frac{\pi D^2}{12}.$$

The remaining part of the area of diaphragm/membrane, equal to difference

$$S - S_{э\phi} = \frac{\pi D^2}{4} - \frac{\pi D^2}{12} = \frac{\pi D^2}{6},$$

in the transmission of force to central support does not participate.

In the given calculation we disregarded the hardness of the material of diaphragm/membrane.

Force of the diaphragm/membrane, which has rigid center. For an increase in the useful force of diaphragm/membrane in its center, establish/install rigid support (center) a with a diameter of d (Fig. 33b).

The calculation of useful force and effective area of the quite membrane/diaphragm fabric is conducted also in this case on the assumption that the membrane/diaphragm fabric possesses the ideal elasticity, and the sagging/deflection of diaphragm/membrane is equal to zero.

Let us isolate on the effective surface of flat/plane membrane/diaphragm ring the elementary circular area/site $dS = 2\pi\rho\, d\rho$, moved away from the center of diaphragm/membrane up to distance ρ. Force from effect of pressure p (counterpressure we disregard) to this surface element

$$dF = dSp = 2\pi\rho p\, d\rho$$

will be transferred to mobile/motile in axial direction rigid center and the motionless washers of the external jamming of

membrane/diaphragm ring in relation, inversely proportional to distances m and n from the places of the jamming of ring to the surface element in question.

In this case the elementary force dF, transferred to the rigid center a,

$$dF_1 = dF\,\frac{n}{m+n} = \frac{R-\rho}{R-r}\,2\pi\rho p\,d\rho,$$

where r and R are a radius of rigid center and the radius of a circle of the jamming of membrane/diaphragm ring in housing.

In accordance with this force F_1, transferred to rigid center from the working (circular) section of membrane/diaphragm fabric (not allowing for the area of rigid center),

$$F_1 = \int_r^R \frac{R-\rho}{R-r}\,2\pi\rho p\,d\rho.$$

$$F_1 = \frac{2\pi p}{R-r}\left[\left(\frac{R^3}{2} - \frac{Rr^2}{2}\right) - \left(\frac{R^3}{3} - \frac{r^3}{3}\right)\right].$$

After integration

$$R = \frac{D}{2} \text{ и } r = \frac{d}{2}$$

After replacing

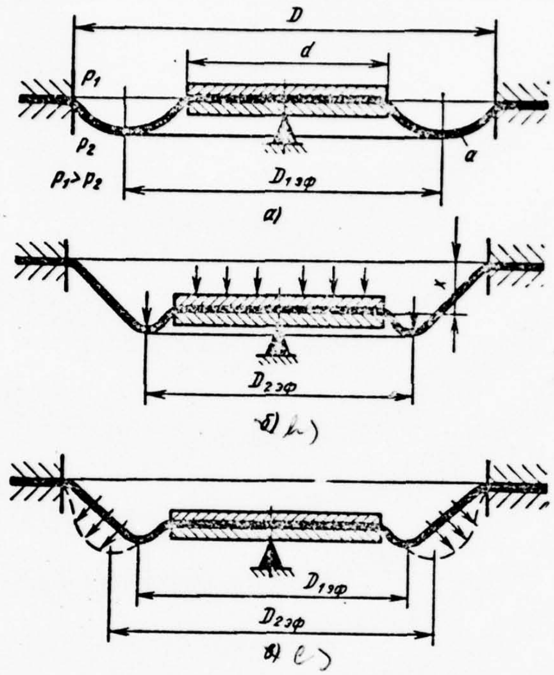
and by converting, we will obtain

$$F_1 = \frac{\pi p}{3}\left(\frac{D^2}{4} + \frac{Dd}{4} - \frac{d^2}{2}\right).$$

Pressure p will also act on rigid center with an area of S_2, developing force

$$F_2 = S_2 p = \frac{\pi d^2}{4} p.$$

Fig. 34. Diagram of the action of membrane/diaphragm actuating mechanism.



Complete force F, developed with diaphragm/membrane under the taken conditions of zero sagging/deflection,

$$F = F_1 + F_2.$$

After substituting values F_1 and F_2 and by converting, we will cbtain

$$F = \frac{\pi p}{12}(D^2 + Dd + d^2). \quad (35)$$

Effect of the displacement/movement of rigid center (sagging/deflection of diaphragm/membrane) to effective area. The given calculation is produced on the assumption that the diaphragm/membrane is located in the mid-position (it has negligible sagging/deflection), i.e., under the assumption of the fact that the plane of the stopping up of diaphragm/membrane in rigid center coincides with the plane of its stopping up in external perimeter (see Fig. 33b and 34a). For this calculated case in the equation of

working force (35) the sagging/deflection of diaphragm/membrane does not enter.

However, the actual working force of diaphragm/membrane changes with its sagging/deflection, i.e., the static characteristic of diaphragm/membrane takes the form

$$F = f(x),$$

where x is sagging/deflection of diaphragm/membrane.

In view of this in moving rigid center from the average (neutral) position, the force, developed by diaphragm/membrane, changes.

The effective area of the aaaaaa of the diaphragm/membrane on which depends develop with it force, is composed, according to foregoing, of two parts:

$$S_{эф} = S_1 + S_2,$$

where S_1 and S_2 it is composed the effective area of the section of the sagging of membrane/diaphragm fabric and rigid center.

Since area S_2 rigid center remains in moving this center in the axial direction of constant, all the changes in the effective area during the displacement of the rigid center of diaphragm/membrane are caused only by a change in the effective area S_1 the section of the sagging of the quite membrane/diaphragm fabric.

The derivation of the equations of the characteristic of diaphragm/membrane taking into account its sagging/deflection requires the examination of complex spatial problem, in view of which in practice they are restricted to the approximate relationships and experimental data. Specifically, the effective area of the $S_{эф}$ of diaphragm/membrane can be calculated according to volume of V the

displaced by it liquids upon transition diaphragm/membrane under the action of load F from position of a, depicted on Fig. as 31 broken lines, in position of b:

$$S_{э\phi} = \frac{V}{x},$$

where x is the course of the washer of diaphragm/membrane, measured according to its axle/axis.

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If we avail by the current force F, developed with diaphragm/membrane, and the pressure of liquid p, then is the current effective area of diaphragm/membrane

$$S_{э\phi} = F/p.$$

The indicated change in the effective area in moving rigid center is caused by a change in the form of the section of the free sagging of membrane/diaphragm fabric. Since the line of deflection of fabric is determined in essence by the mechanical properties of the material from which it is made, this line for different materials will be under otherwise equal conditions different.

Virtually they consider that the effective area of diaphragm/membrane is determined by the area of circumference by the diameter of the $D_{эф}$, carried out from the apex of curve of the sagging (sagging/deflection) of membrane/diaphragm fabric (Fig. 34a):

$$S_{эф} = \frac{\pi D^2_{эф}}{4}.$$

Experiments show, that corrugated diaphragms (see Fig. 30) they retain the linearity of response of $S_{эф}$ = f (x) within the more considerable limits of course, than planes. The most rational form of corrugation it is plate (see Fig. 30c), diaphragm/membranes of this form from the rubberized twill fabrics they possess good linear characteristic and small hysteresis.

With the small thickness of fabric (0.25-0.3 mm) hysteresis of these diaphragm/membranes does not exceed 2-3o/o of range of a change in the effective area.

In the free position of the rigid center of diaphragm/membrane,

the effective area does not depend on the form of corrugation and of pressure differential and is determined (not allowing for the drawing of material) according to equation (35).

During the displacement of rigid center from free position (Fig. 34b) is observed the change in the effective area, caused by the pressure which will be the greater, the greater the displacement x. From Fig. 34b it follows that maximally possible displacement x of rigid center from free position (not allowing for the drawing of the material of fabric

$$x_{max} = \pm \sqrt{l^2 - \left(\frac{D-d}{2}\right)^2},$$

where l - the arc length of corrugation a (see Fig. 34a).

At this value of x_{max} effective area the diaphragm/membranes and respectively force in its rigid center will be equal to zero. It is obvious, during the displacement/movements of rigid center, close to that which was indicated, calculation according to given formula

(35) gives considerable errors.

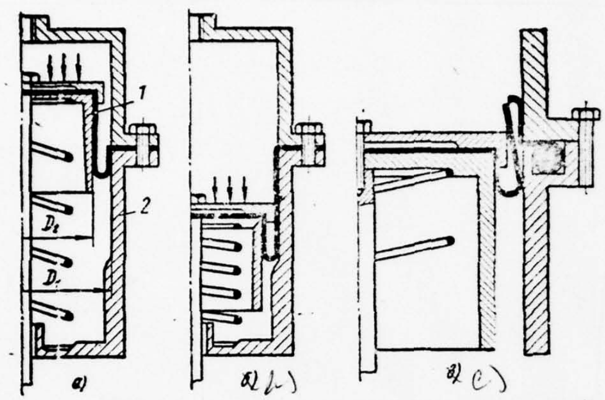
Decrease during the displacement of the rigid center of the force, developed with diaphragm/membrane, is caused by a change in the effective area of elastic ring, and also fact that in proportion to the increase in the sagging/deflection of membrane/diaphragm fabric it elastic is transformed, on what is expend/consumed part of the force, developed with the pressure of medium, appropriate vertical component tensile stress of membrane/diaphragm fabric.

This first of all concerns flat diameters (see Fig. 33), in view of which during their calculation according to formula (35) is introduced correction factor of k. In this case, the calculation formula takes the form

$$S_\partial = k \frac{\pi}{12} (D^2 + Dd + d^2),$$

where k = $\frac{S_\partial}{S}$ - coefficient; here S_∂ S - the real and calculated effective area of diaphragm/membrane.

Fig. 35. Membrane/diaphragm separators with large course.



Coefficient k depends on the course of diaphragm/membrane, properties of material and ratio d/D. For d/D = 0.5 and the rubberized lastic this coefficient can be accepted with small course equal to 0.97. For the widespread in practice ratio d/D = 0.6 value k ~~ 1.

Dependence of the effective area of diaphragm/membrane on the pressure differential. The second in significance reason for a change in the effective area and disturbance/breakdown of the stability of the characteristics of diaphragm/membranes is the action of pressure differential. The dependence of the effective area of diaphragm/membranes on the pressure differential in essence is caused by the strain (drawing) of memirane/diaphragm fabric under the action of the tensile stresses, determined by this jump/drop.

With the loading of elastic diaphragm/memiranes the pressure differential they acquire noticeable hardness as a result of a change in the effective area in moving rigid center.

Figure 34c shows the diagram of the caved in flexible diaphragm/membrane, which is located under certain low pressure (solid line). Its effective area, determined for this position of

center by the diameter of $D_{1эф}$,

$$S'_{эф} = \frac{\pi D_{1эф}^2}{4}.$$

In a pressure increase and the constant/invariable position of rigid center, the elastic surface of diaphragm/membrane will take the form, shown by broken line, whereupon its effective diameter will increase and it will become equal $D_{2эф}$. In accordance with this, the effective area of diaphragm/membrane will increase

$$S''_{эф} = \frac{\pi D_{2эф}^2}{4}.$$

For a change in the effective area during a change in the pressure differential, affects also the drawing of membrane/diaphragm fabric under the action of tensile stresses, as a result of which changes the length of corrugation, which depends on the mechanical properties of material and of pressure differential.

Are applied also actuating mechanisms with the flexible separators, which allow/assume the considerable displacement/movements of rigid center (Fig. 35). In moving rigid center (piston) 1 in the direction of the effect of pressure of liquid (Fig. 35a) diaphragm/membrane is bendd, being rolled from the walls of piston 1 to the walls of cylinder 2, to which it tightly is adjusted by the pressure of liquid (Fig. 35b).

The transferred to rigid center force is composed of force F_1, developed with pressure p of liquid on the elastic part of the diaphragm/membrane, and forces F_2 of pressure on its rigid center. Under the condition infinitesimal thickness of the fabric of diaphragm/membrane the sectional area of the surface of bend

$$S_1 = \frac{\pi}{4}(L_1^2 - D_2^2).$$

where D_1 and D_2 are diameters of cylinder and rigid center (piston).

Taking into account that effective part will be the half of this area, force

$$F_1 = \frac{\pi}{8} p (D_1^2 - D_2^2).$$

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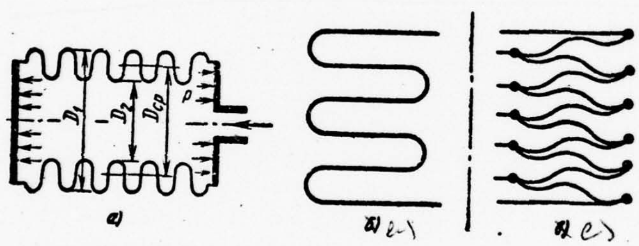

Fig. 36. Diagram of metallic bellows.

Force F_2, developed by piston,

$$F_2 = \frac{\pi}{4} D_2^2.$$

Total force, which is transferred to rigid center,

$$F = F_1 + F_2 = \frac{\pi}{8} (D_1^2 + D_2^2).$$

Similar rubber-fabric diaphragm/membranes do not allow/assume bilateral loading, since are formed the supplementary bends (Fig. 35c), as a result of which the diaphragm/membrane will be destroyed.

Bellows.

In automatic hydraulic-pneumatic equipment are applied also actuating mechanisms with separator in the form of cylindrical bellows (Fig. 35a). Bellows manufacture from metals and only for a work at small pressures - from nonmetallic materials (rubber and different plastics).

Metallic bellows are one- and multilayer (to five layers), whereupon multilayer bellows allow/assume with the same overall thickness, as single-layer, and with the same size/dimensions considerably larger course with identical load. Permissible pressure for nonmetallic bellows to 2-3 kgf/cm², for the single-layer metallic bellows of small diameters to 30 kgf/cm² and large (>150 mm) - to 2 kgf/cm². Multilayer bellows made of the stainless steel are applied for operating pressures to 150 kgf/cm². The application/use of these bellows has special advantages under conditions of the low and high temperatures whose value is limited by the material from which is made the bellows.

The life of metallic bellows is characterizedby the total number of courses of the assigned magnitude to the destruction of any of its waves, this number of courses depending on size/dimension and frequency of the strains whose increase lowers the life of bellows. The total change in the length (course) of bellows consists of elongation (elongation) and compression. The recommended maximum displacement/movement of metallic bellows is 25o/o of its free length from which 15o/o is abstract/removed for compression 10o/o for elongation. If necessary of providing a large number of courses, a change in the length of bellows must not exceed 10o/o. The permissible axial displacement of bellows from rubber can be depending on the size/dimension of corrugations taken as 50o/o of its complete length in free state into each side.

Bellows more preferable to load with external pressure, the permissible pressure by this case exceeding pressure with internal loading on 25-30o/o.

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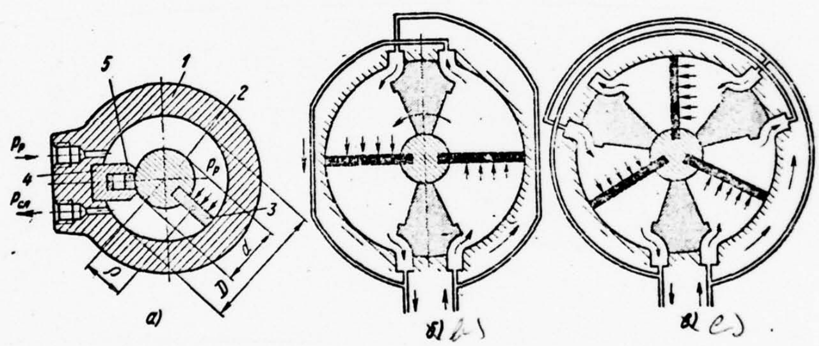

Fig. 37. Moment hydraulic cylinders.

As the effective diameter of bellows, approximately is accepted the mean diameter of the D_{cp} of corrugations (see Fig. 36a), in accordance with which the force, developed with bellows during the action of the internal pressure of liquid, can be approximately, disregarding the effect of the hardness of the material of bellows, is calculated as product of pressure p on the area of circle with the mean diameter of the D_{cp}:

$$P = Fp = \frac{\pi}{4} pD_{cp}^2,$$

where $F = \frac{\pi D_{cp}^2}{4}$ is a useful (effective) area of bellows;

$D_{cp} = \frac{D_1 + D_2}{2}$ - the mean diameter of the corrugations of bellows, here D_1 and D_2 are external and internalizations the diameter of corrugation.

Virtually the relation of external $R_н$ and internal $R_в$ of radii comprises

$$\frac{R_н}{R_в} \leqslant 2.$$

The precision determination of effective diameter (the effective area of bellows) can be produced only by measurement. After assuming that the pressure differential Δp, which acts from inside, lengthens bellows to value Δx, and after balancing this jump/drop by external force ΔF, they calculate the effective area of bellows according to formula

$$S=\frac{\Delta F}{\Delta p}.$$

Bellows in essence manufacture by two methods: by the rolling-out of thin-walled seamless pipe (Fig. 36b) and by welding on the end/faces of separate shaped rings (Fig. 36c). During the production of bellows in an welded manner it is represented possible to obtain the corrugations of any height/altitude, whereas the height/altitude of the corrugations, prepared by rolling-out, is limited by the possibility of the drawing of material. Therefore welded bellows allow/assume at the same length higher reduction (course), than bellows from ducts. The possibility of an increase in the reduction is caused also by the fact that to corrugations in these bellows it is possible to give this form in order that they would enter one in another.

Moment hydraulic cylinders (Turners)

For the recurrent-rotary motions of the given nodes to an angle less than 360°, are applied the moment hydraulic cylinder or the turner, which is volumetric hydraulic engine with the recurrent-rotary relative to housing motion of operating unit (Fig. 37).

Moment hydraulic cylinder consists of the housing of 1 and rotary rotor, which is bushing 2, carrying plate (blade/vane) 3. The annular passage between the inner cylinder face and the rotor is divided sealing cross connection 4 with the springy clamping/tightening to the rotor of sealing cell/element 5.

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During the supply of liquid under the pressure of p_p into the upper channel (Fig. 37a shows by arrow/pointer) plate 3 with bushing 2 will be turned clockwise. The shaft position of cylinder with one working plate usually does not exceed 270-280°.

The calculated torsional moment M on the shaft of the hydraulic cylinder in question with one plate is equal to the product of force P, developed the pressure differential of the liquid of $\Delta p = p_p - p_{c\text{л}}$ on effective area F of plate, on arm ρ the application/appendix of this force (distance from rotational axis to the center of pressure of the effective area of plate, see Fig. 37a):

$$M = P\rho = \Delta p F \rho.$$

Figure 37a shows that the effective area of plate

$$F = \frac{D-d}{2} b$$

and the arm of application/appendix try

$$\rho = \frac{D}{2} - \frac{D-d}{4} = \frac{D+d}{4}.$$

In accordance with this the calculated torsional moment

$$M = P\rho = \Delta p F \rho = \frac{\Delta p\,(D-d)\,b}{2} \cdot \frac{D+d}{4} = \frac{\Delta p b}{8}\,(D^2-d^2),$$

where the $\Delta p = p_p - p_{c\text{д}}$ - the pressure differential between of working (p_p) and drainage $(p_{c\text{д}})$ by the cavities of cylinder; D and d are cylinder bore and the diameter of rotor; b - the width of plate along the axis of cylinder (length of cylinder).

Angular velocity ω the shaft of cylinder will be determined from

— the condition of the equality of the fluid flow rate Q and of the volume, described by plate per unit time:

$$Q = v_{ok}F,$$

where the v_{ok} it will be determined the peripheral speed of the center of pressure of plate.

After substituting values

$$F = \frac{D-d}{2}b \text{ and } v_{ok} = \omega\rho = \omega\frac{D+d}{4},$$

we will obtain

$$Q = \omega\frac{D+d}{4}\cdot\frac{D-d}{2}b = \frac{\omega b}{8}(D^2 - d^2).$$

From this expression we find

$$\omega = \frac{8Q}{(D^2 - d^2)\, b}.$$

The actual torque/moment of M_ϕ and the angular velocity of ω_ϕ will be less than the calculated in connection with the presence losses of friction and hydraulic slipes, characterized mechanical $\eta_{мех}$ and volumetric $\eta_{об}$ the efficiency of the hydraulic cylinder:

$$M_{\phi}=\frac{b\,\Delta p}{8}(D^2-d^2)\,\eta_{мех};$$

$$\omega_{\phi}=\frac{8Q\eta_{об}}{(D^2-d^2)\,b}.$$

Are applied also the multiplate moment hydraulic cylinders (Fig. 37b and c), which make it possible to increase the torsional moment; however, angle of rotation in this case it decreases.

Torque/moment M and angular velocity ω the multiplate hydraulic cylinder:

$$M=\frac{z\,\Delta pb}{8}(D^2-d^2);\quad \omega=\frac{8Q}{zb\,(D^2-d^2)},$$

where z is a number of plates (pistons).

For a hydraulic cylinder with one plate, the angle of rotation of output shaft can reach 280°, with two -140°. Hydraulic cylinders are discharged in essence to the pressure differential 200-300 kgf/cm^2.

~~end section~~

TRANSFORMATION OF RECTILINEAR MOTION INTO ROTARY.

The rotary motion of shaft can be also realized by transformation with the aid of the different mechanical cell/elements of the rectilinear forward motion of piston.

Specifically, in the machine tools widely is applied the swivel gear with gear and toothed rack (Fig. 38a). The Teeth of rack usually are cut on the plunger (piston) of cylinder.

torque and rotary motion are transferred to the output shaft through the gear, meshed with rack. Depending on the length of rack, is obtained the angle of rotation from 90 to 360° and above.

The calculated torsional moment of this pivot

$$M = \frac{pi'd}{2},$$

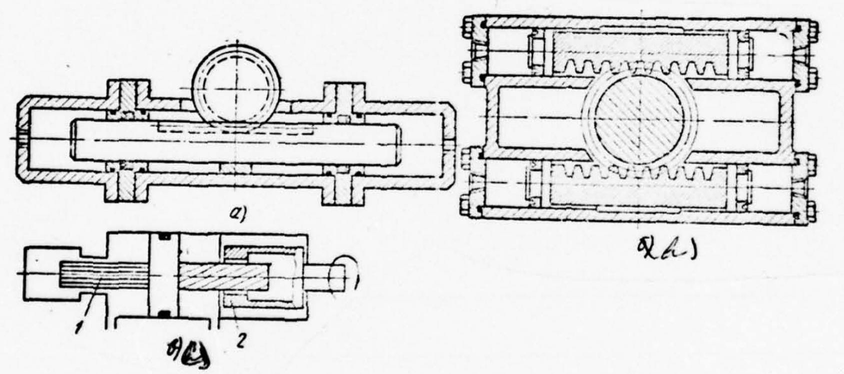

Fig. 38. Diagrams of mechanisms for the transformation of rectilinear motion into rotary.

where p is pressure of liquid (usually 200 kgf/cm²); $F = \pi D^2/4$ - piston clearance; D and d of the diameter of piston and pitch circle of gear.

Fluid flow rate

$$Q = \frac{Fd\gamma}{360},$$

where γ is an angle of rotation of output shaft.

In the transmission of the large torsional moments, are applied the pivots with two cylinders whose stock/rods are racks (Fig. 38b).

For the transformation of the forward motion of piston into rotary, frequently is applied spline- helical mechanism (Fig. 38c). Right stock/rod 1 is executed in the form or screw/propeller with angle of ascent $\alpha = 70°$, that enters in to gallery 2, that is exit cylinder and accomplishing only rotary motions. Piston is fixed from rotary motions with the aid of splined joint of its left stock/rod with the splined bushing of actuating cylinder.

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In this case during the axial displacements of piston, nut 2 and, consequently, also exit cylinder accomplish rotary motions.

Displacement h of piston and angle ϕ the rotation of nut are connected with single thread by expressions

$$h = \frac{\varphi}{2\pi} s = \frac{\pi d \operatorname{tg} \alpha \cdot \varphi}{2\pi} = \frac{d \operatorname{tg} \alpha \cdot \varphi}{2},$$

where s and d - the space and the mean diameter of the cutting of screw/propeller; α and φ are angles of ascent of thread and rotation of nut.

The angular velocity of exit cylinder

$$\omega = \frac{\pi v}{30 s} \quad \text{rad/s},$$

where v is a linear piston speed.

The torsional moment M fluid flow rate Q of this drive

$$M = pF \operatorname{tg}(\alpha + \mu) r;$$
$$Q = \frac{F \cdot 2r\pi \operatorname{tg} \alpha \cdot \varphi}{360},$$

where p is pressure of liquid; r - the mean radius of thread of screw; μ - the angle of friction F - piston clearance.

Mechanisms of such type are designed for pressures to 200 kgf/cm^2.

Mechanisms (hydraulic motors) of rotary action.

As the actuating mechanisms (hydraulic engines) of rotary action serve the hydraulic machines (hydraulic motors) of the same design concept, as pumps, but with the slide-valve (valveless) distribution of liquid.

The power supply of hydraulic motor is realized either from common/general/total hydraulic system or by an individual pump.

System consisting of pump and hydraulic engine of rotary action (see Fig. 3b and c), is volumetric hydraulic drive which structurally can be executed in the form of the integral unit (machine), which

includes pump and the motor of one construction or the other (inseparable performance), or in the form separate pump and hydraulic motor, connected with conduit/manifolds (separate performance).

Reversing the direction of the motion (rotation/revolution) of hydraulic motor is realized either with the aid of distributor (see Fig. 3b), or by reversing the direction of the feed of pump (see Fig. 3c).

The control of the output shaft of hydraulic motor realizes by a change in the amount of entering it liquid with the aid of throttle governor (see Fig. 3b) or by a change in the pump displacement or hydraulic motor (volumetric control, see Fig. 3c). The theoretical power of N_T and the torsional moment of M_T on the output shaft of hydraulic motor are connected by relations

$$N_T = \Delta p Q_T = \Delta p q n;$$
$$M_T = \frac{N_T}{2\pi n} = \frac{\Delta p q}{2\pi} = 0{,}159\,\Delta p q, \qquad (36)$$

where $\Delta p = p_1 - p_2$ - the pressure differential in hydraulic motor; here p_1 and p_2 are pressure of liquid of the inlet into hydraulic motor and at output/yield from it; q is the working volume of hydraulic motor; n - the frequency of the rotation/revolution of the shaft of hydraulic motor.

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When using extrasystemic ones theoretical the power of N_T h.p. and the torsional moment of M_T kg•m are expressed by equations

$$N_T = \frac{\Delta p Q_T}{45 \cdot 10^4};$$
$$M_T = 716{,}2\,\frac{N_T}{n},$$

where Δp - the pressure differential in kgf/cm²; $Q_T = qn$ are the design capacity in cm³min; n - rotation frequency in r/min.

MECHANICAL LOSSES AND EFFICIENCIES.

The energy conversion (hydraulic into mechanical) in hydraulic motor is provided by the motion of working cell/elements, which is accompanied by energy losses (power) for friction of mechanical features and liquid.

The indicated power losses are determined a the difference between the theoretical (indicator) power and with a power on the shaft of hydraulic motor.

In accordance with this the actual torque developed with hydraulic motor,

$$M_{эф} = M_T - \Delta M,$$

where the M_T - theoretical (indicator) torsional moment of hydraulic motor (not allowing for losses for mechanical friction and liquid resistance); ΔM - the loss of torque/moment.

Mechanical power losses can be to also determined as the difference between the theoretical power of N_T and the effective power of $N_{эф}$ on the shaft:

$$\Delta N = N_T - N_{эф}.$$

The losses in question are characterized by the mechanical efficiency of the hydraulic motor of the $\eta_{мех}$, which is equal to the ratio of the effective $N_{эф}$ of power on its shaft to of the theoretical N_T of the power:

$$\eta_{мех} = \frac{N_{эф}}{N_T} = 1 - \frac{\Delta N}{N_T}$$

or

$$\eta_{мех} = \frac{M_{эф}}{M_T} = 1 - \frac{\Delta M}{M_T}.$$

For the hydraulic motors of average power (10-100 h.p.) it is possible to accept

$$\eta_{мех} = 0{,}92 \div 0{,}96.$$

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Chapter III

Devices (apparatuses) of distribution and control.

Important value for the reliable work of hydraulic systems has the rational selection of the governors, which ensure the execution of logic functions on the realization of the assigned sequence of the action of the actuating mechanisms of hydraulic system. Most important of these functions is the steering and fluid flow rate, for which is applied the different control equipment of flow rate and pressure, change in the direction of the flow of liquid, connection/inclusion and disconnection of separate/individual actuating mechanisms and separate/individual sections of conduit/manifold etc.

DISTRIBUTORS OF LIQUID.

Distributor (distributor) is intended for a control of the flow of working liquid. With the aid of distributors is provided the direction of working liquid to the appropriate actuating mechanism, and also is realized the reverse of hydraulic mechanisms.

On constructive execution the distributors divide in essence into slide-valve, crane and valve types. In the first type liquid distribution is realized with the aid of the axial displacement of cylindrical or flat/plane distributive cell/element, in the second - by means of the rotation of cock plug and into the third - by means of consecutive discovery/opening and closing of working (expenditure) windows with the aid of valves (gates).

By working window is understood the flow area of the hydraulic apparatus in which directly occurs a change in the flow parameter of working fluid.

Slide-valve distributors.

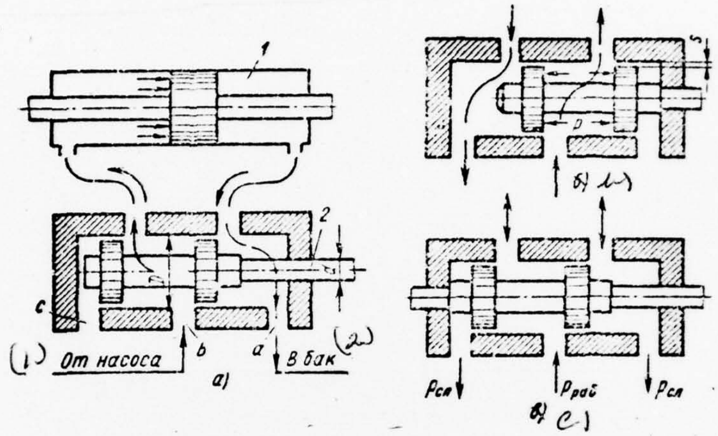
The operating unit of distributors of this type is the being moved in axial direction in bushing (case) cylindrical plunger on which executed corresponding amount of circular flows. The supply and the diversion/tap of liquid is conducted through the windows of power supply in the bushing (housing) of distributor and the corresponding annular grooves of its plunger.

In the amount of connected outlines (channels of power supply), on are distinguished four-line (fourway), three-line and bilinear distributors.

Figure 39a and b shows the diagram of the fourway slide-valve distributor, intended for a control of the bilateral motion of the hydraulic engine, realized by means of the supply of the entering from pump liquid under pressure into one of the two cavities of hydraulic engine with its simultaneous diversion/tap of opposite cavity into reservoir.

Fig. 39. Types of distribution valves.

Key: (1). From pump. (2). In tank.



Liquid from pump will be fed to channel b (Fig. 39a), from which for this position of plunger 2 it enters the left cavity of hydraulic engine 1; simultaneously with this the second (nonoperative) cavity of hydraulic engine is connected with channel a, which drives into tank. In moving plunger 2 to the right (Fig. 39b) the directions of fluid flows change.

The major advantage of slide-valve distributors is the fact that their plungers are balanced from the axial static forces of the operating pressure of liquid, since this pressure acts on the bands of plunger in opposite directions.

For the balancing of the forces of pressure of the $p_{сл}$ of liquid in drainage line (in channels a and c) the plunger of the valve, presented in Fig. 39c, is equipped from left side by false shank. In the absence of shank (Fig. 39a) the pressure of $p_{сл}$ in the drainage line with which are connected channels a and c, it will affect the unbalanced area of plunger

$$f=\frac{\pi d^2}{4},$$

where d is a diameter of shank, attempting to displace plunger to the right.

This unbalanced force of the pressure of liquid

$$P = p_{cA} f = p_{cA} \frac{\pi d^2}{4}.$$

For this purpose the plunger of valve (Fig. 40) is equipped besides the basic bands c and d by the supplementary bands a and b which the equilibrants of drainage pressure.

The balancing of the plunger of valve from drainage pressure can be reached also by applying a three-band valve, executed by the diagram, presented in Fig. 41.

Are applied also three-way less frequently thinner two-pass valves, whereupon the latter are actually pet cocks (valve/gates). Three-way valves (Fig. 42a) they apply in essence when the window of the power supply of hydraulic engine must be consecutively connected with the source of pressure (with pump) and with reservoir, i.e., they are applied in the hydraulic engines of one-sided action (see Fig. 26c). NP Page 69.

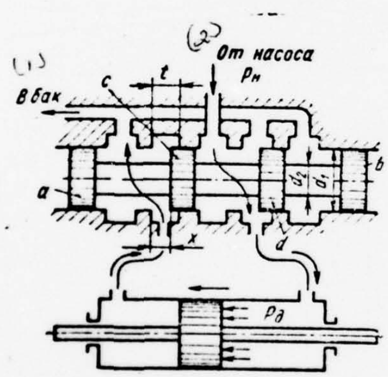


Fig. 40. Fourway slide-valve distributor.

Key: (1). In tank. (2). From pump.

Fig. 41. Distribution valve.

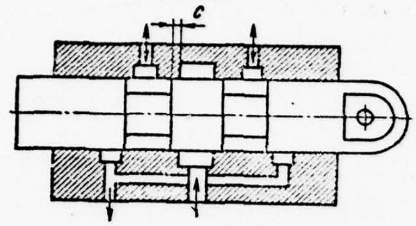
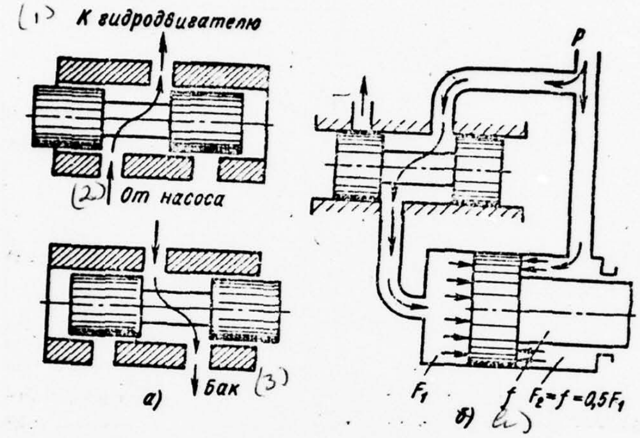


Fig. 42. Three-way slide-valve distributor.

Key: (1). To hydraulic engine. (2). From pump. (3). Tank.



However, in certain cases they are applied also in diagrams with bilateral hydraulic engine. A similar diagram with the three-way valve, which allow/assumes a change in the direction of the motion of hydraulic engine, is represented in Fig. 42b. In this diagram is applied the actuating cylinder of the double action in which effective piston clearance stock/rod's on the part half piston clearance from opposite side (see Fig. 26d). In the position of plunger, shown in Fig. 42b, the liquid enters from power supply simultaneously both into left and into right the cavity of cylinder, as a result of which piston is moved to the right. The speed of the piston stroke v and developed by it force P are determined depending on the supply of liquid by supply of power Q from expressions

$$v=\frac{Q}{F-f};\quad P=p(F-f),$$

where F - the area of cylinder.

Under condition f = F/2, we will obtain

$$v=\frac{2Q}{F};\quad P=\frac{pF}{2}.$$

During the connection of the left cavity of cylinder with gutter and right with power supply the piston will be moved to the left with speed

$$v = \frac{Q}{F-f} = \frac{2Q}{F},$$

developing in this case force

$$P = p(F - f) = \frac{pF}{2}.$$

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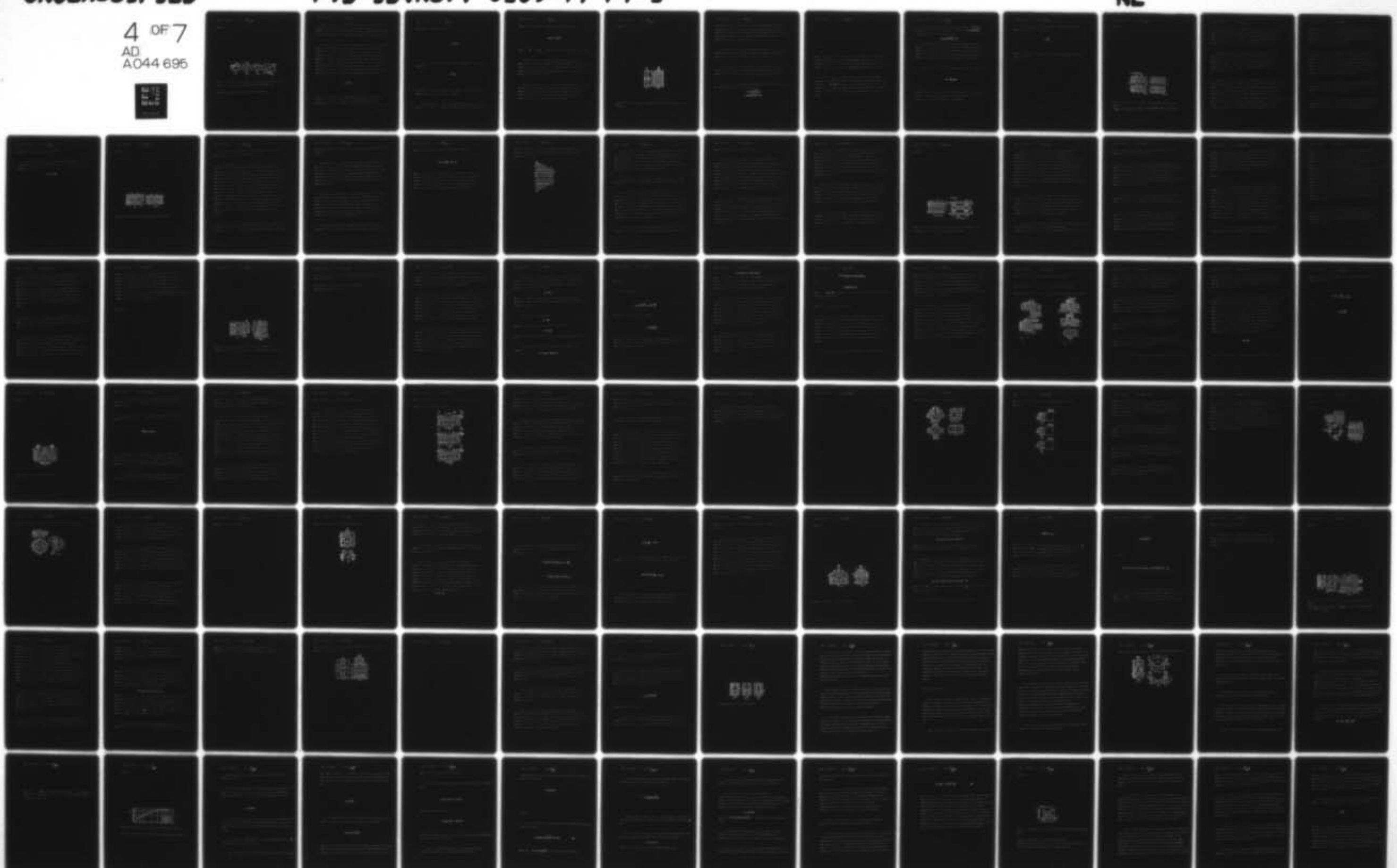

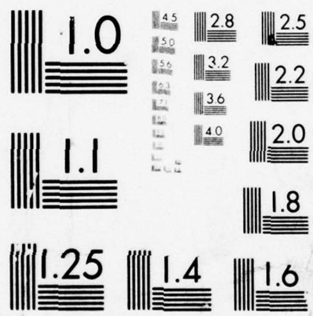

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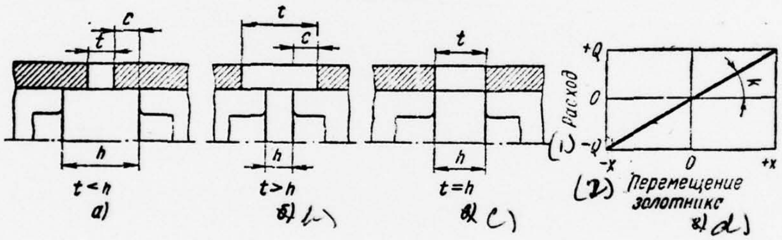

Fig. 43. Diagrams of the overlaps of windows in distribution valves (a-c) and the static characteristic of ideal valve (d).

Key: (1). Flow rate. (2). Displacement of valve.

By the number of fixed positions of plunger, are distinguished two- and three-position valves. If the plunger of valve is not detained in the mid-position, then this valve is called two-position; if it is detained with the aid of any devices, then three-position.

Overlap of slide valve ports. For many cases of applying valves and, in particular, for the hydraulic servo systems the important parameter is the overlap with plunger in its average/mean position of expenditure windows (Fig. 43). Are distinguished distributors with positive (Figs. 43a and 41) and negative (Fig. 43b) overlap. More rarely are applied valves with zero overlap (Fig. 43c). In first type valves (Fig. 43a) width h of the working band of plunger sweep t of the passage opening of the housing of valve for the duct of liquid; therefore to windows overlaps the appropriate window at length

$$c=\frac{h-t}{2};$$

in second type valves (Fig. 43b) width h of working band less than the width t of passage opening, as a result of which in the mid-position of plunger valve along both sides of his band is formed

the initial clearance, equal to

$$c=\frac{h\cdot -t}{?}.$$

Since under condition $h < t$ overlap c, calculated according to expression

$$c=\frac{h-t}{2},$$

is negative, the similar overlap of windows by the sealing bands of valve usually is called negative overlap.

In hydraulic system with this valve in the cavities of actuating cylinder, they will be establish/installed in the mid-position of the

plunger of the valve of pressure p_1 and p_2 (Fig. 44):

$$p_1 = p_2 = \frac{p_{н} + p_{сл}}{2},$$

where $p_{н}$ and $p_{сл}$ - pressure in the delivery lines and gutter.

During the displacement of plunger to any side from free position, the indicated equality of pressures into the cavities of cylinder will be disrupted, as a result of which the piston of actuating cylinder with the known difference of pressures will be moved to the appropriate side.

Obvious that if there is no load and frictional forces in power servomotor (at output/yield), then any disturbance/breakdown of the equality of pressures p_1 and p_2, caused as small as desired displacement of the plunger of valve relative to its average position, will cause the motion of this engine.

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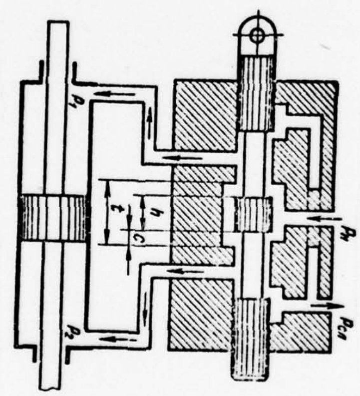

Fig. 44. The design diagram of valve with the negative overlap of windows.

In actuality, for the overcoming of load and frictional forces of output/yield in the cavities of actuating cylinder must be the determined pressure differential, and consequently, the valve of this diagram will have a dead zone which increases with an increase in the initial clearance (negative overlap).

A shortcoming of valves with negative overlap is the loss of the liquid, which overflows through expenditure windows to gutter with the average and the close to it positions of plunger.

A deficiency/lack in the valves with negative overlap is the loss of the liquid, which overflows through expenditure windows to gutter with the average and the close to it positions of plunger.

In practice the hydraulic characteristic of valve (see Fig. 40) is determined according to equation (20)

$$Q = \mu f \sqrt{\frac{2}{\rho} \Delta p} = \\ = \mu \pi d_1 x \sqrt{\frac{2}{\rho} (p_н - p_д)},$$

where $f = \pi d_1 x$ - the sectional area of the passage slot of valve; x is discovery/opening valve (size/dimension of slot); d_1 - the diameter of the plunger of valve; $p_н$ and $p_д$ - pressure at inlet (power supply) and output/yield (pressure of the load or engine).

The given equation shows that with an increase in the $p_д$ (pressure of the load of engine) the flow rate through the valve with x and the $p_н = \text{const}$ decreases. This phenomenon (throttling effect) it lowers the hardness of the mechanical characteristic of hydraulic drive and causes the slip of hydraulic engine under the action of load.

At the zero pressure of the load of the engine of $(p_\partial = 0)$, the flow equation through the valve the where of $k = \mu\pi d_1\sqrt{\frac{2}{\rho}p_н}$ - amplification factor in flow rate.

$$Q = \mu\pi d_1 x\sqrt{\frac{2}{\rho}p_н} = kx,$$

Consequently, in ideal valve occurs the linear dependence of flow rate Q on control signal (displacement of valve) x (see Fig. 43d). This property of valves has extremely important practical value, which caused their wide application/use, in particular in the slave/servo hydraulic drive and in the systems of automation.

The value of the coefficient of flow rate μ can be accepted with

$$\mathrm{Re} = \frac{2xu}{\nu} \geqslant 200$$

constant (μ = const). For mineral oils and valves with the sharp edges of slots this coefficient can be taken as for the indicated condition μ = 0.62-0.65. With Re < 200 μ ≈ 0.5.

The sectional area of the channels by which are provided the given speed of flow and flow rate,

$$f=\frac{Q}{u},$$

where Q is provided fluid flow rate; u is the given speed of fluid flow in channel.

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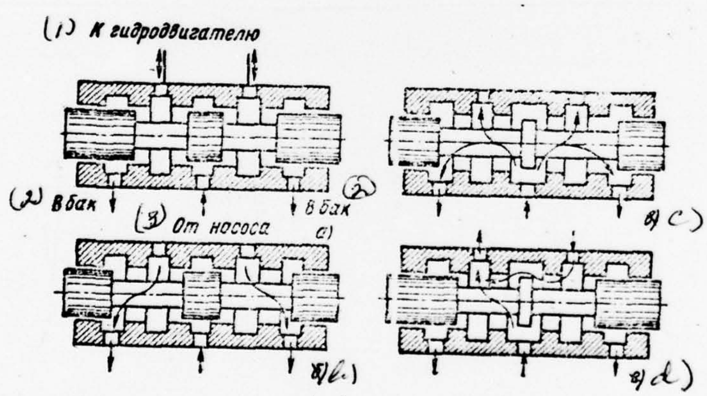

Fig. 45/ diagram of the internal communications of distribution valves.

Key: (1) To hydraulic engine; (2) In tank; (3) From pump.

The rate of flow of the liquid in the channels of the housing of valve and in the annular grooves of plunger, which determines the flow resistance, usually selects for a decrease in the dimension of valve 3-5 times higher than the rate of liquid in supply lines. Virtually the rate of liquid is accepted to 10-15 m/s.

Third type valves (see Fig. 43c) with zero overlap (t = h) are applied when it is required in order that during any small displacement of plunger from the mid-position would be formed expenditure slot. The similar cases include the hydraulic servo systems of the copying machines in which is required the high accuracy/precision of tracking.

Depending on the construction of valve with the positive overlap of the window of power supply, the working cavities of hydraulic engine in the mid-position of plunger either are cut off (blocked), or they are connected with reservoir. Figure 45 shows the possible connections of the channels of power supply in the mid-position of plunger. In the diagram with positive overlap, presented in Fig. 45a, are blocked (are overlapped) all channels of valve; in the diagram, shown in Fig. 45b, is blocked only the channel of power supply, channels, connected with the cavities of hydraulic engine, are

connected with tank; in the diagram with negative overlap, presented in Fig. 45c, with tank are connected all channels of slide valve, thanks to which are provided the discharging of pump and the "floating" (possibility of free displacement under the action of external forces) of actuating element.

Figure 45d depicts the diagram of valve with the connection of the cavities of actuating cylinder. In the mid-position of the plunger of this valve, both cavities of cylinder are connected between themselves and the pump. Therefore, for example, into the cavity of cylinder with one-sided stock/rod, opposite to stock/rod, will enter both liquid, applied by pump and the liquid, displaced from the rod cavity of cylinder (see Fig. 28).

Hydraulic characteristics of valve. They are determined by its hydraulic friction Δp, which depends on the design features of the concrete/specific/actual copy of valve.

The data of studies show that as a result of the perturbation action of rotations, and also of contractions and flow expansions of liquid on slide-valve distributors is predominantly turbulent,

whereupon critical number is Re = 100-200.

Hydraulic characteristics of distributors sufficiently accurately are expressed by parabola and can be determined in the general case by dependence

$$\Delta p = \zeta \frac{\rho}{2} \frac{Q^2}{f^2}.$$

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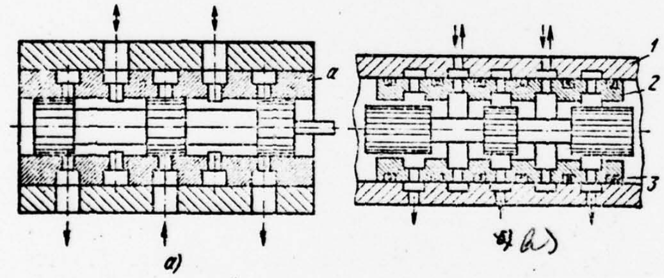


Fig. 46. Slide-valve distributor with floating hub.

The value of drag coefficient ζ depending on the number of rotations of flow varies within limits ζ = 3-5.

The size/dimensions of valve are determined in essence by the flow rate and the permissible rate of flow of the liquid in its channels, which, in turn, depends on the designation/purpose of valve and operating pressure in hydraulic system. The passage channels of valve are selected taking into account the provision for the required fluid flow rate during the permissible flow resistance of liquid; in this case one should approach in order that the piston stroke of valve would be minimum. For this purpose, the supply of liquid into the chambers of valve usually is conducted through circular (circular) annular grooves with a width of t (see Fig. 40). Therefore they reach the maximum value of the perimeter of passage opening in circumference ($w = \pi d_1$) and its areas $f = \pi d_1 x$, where d_1 and x they reach the diameter of plunger and the displacement of the plunger of valve relative to cutoff edges (discovery/opening expenditure window).

Diameter D_2 of the neck of plunger must be so that provided for required flow area, formed with this neck and the internal surface of opening/aperture in the housing of valve [$\pi d_1 t$:mg $(d^2_1$ is $d^2_2) \pi/4$],

and simultaneously with this was preserved the required hardness of plunger.

For providing an airtightness the minimum diametral clearance in slide-valve vapor plunger - bushing usually make equal to 4-10 μm. The surface hardness of the parts of the sliding pair must be highest possible (HRC 60-62). For this provision apply interchangeable (inserted) bushings (case) a (Fig. 46a), which usually are begun pressing into the housing of valve before the finishing treatment of effective surface.

When selecting clearances in slide-valve vapor it is necessary also to consider the temperature expansion of the materials from which are made the parts of plunger pair, in order that removed the jamming of plunger during temperature changes. Virtually for a prevention during a change in the temperature of jamming or formation/education of ample clearances plungers and a pivot valve bushing, must be made from uniform material.

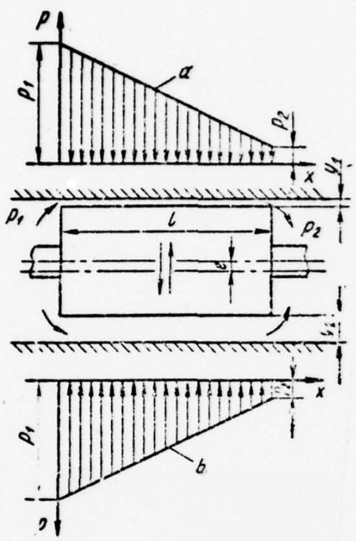
When case (housing) and the plunger of valve are made from materials with the different temperature coefficients of expansion, a

change of the clearance during temperature variations

$$\delta = \delta_0 + \frac{d}{2} (\beta - \alpha)(t - t_0),$$

where t_0 and δ_0 is made the initial temperature of distributor and clearance at this temperature; d is a diameter of the plunger of valve; t and δ are the observed temperature of distributor and clearance at this temperature; β and α are coefficients of the temperature expansion of housing (case) and of plunger.

Fig. 47. Diagram of pressure distribution in the radial clearance of sleeve valve during the bussing arrangement of the axle/axes of plunger and bushing.



For the elimination of the possibility of the jamming of plunger as a result of the temperature strain of the housing in which is placed the case, in particular, if housing is made from nonferrous metal, bushing 2 is placed in the bore of housing 1 freely (with small clearance 0.05 mm); the airtightness of connection in this case is reached with the aid of ferrules 3 (Fig. 46b).

Forces, which act on the plunger of valve. To plunger of valve in essence act the frictional forces and flow forces of fluid flow.

From the examined diagrams of valve, it follows that the forces of pressure of liquid on part ideal the vapors with absolute cylindricity and the high quality of surface treatment are balanced both in axial and in radial direction, but surfaces are balanced both in axial and in radial direction, but the sliding tracks of plunger are divided by the boundary layer of liquid. Consequently, friction of the plunger of this ideal pair depends only on the rate of its displacement and viscosity of liquid.

However, experiment shows that friction of the plunger of real pair depends on the pressure of liquid and on the correctness of the

geometric forms of plunger and bushing.

Friction of plunger appears as a result of the nonuniform distribution of the pressure of liquid in radial radial clearance δ, formed by plunger and case, in consequence of which appears the unbalanced radial force, which adjusts plunger to one wall of case. This lack of balance appears as a result of the disturbance/breakdown of the parallelism of the generatrices of radial slot, and also as a result of avalanches and the other manufacturing defects of these parts. With the correct geometric form of the parts of plunger pair and parallelism of their axle/axes, radial forces are balanced.

Let us examine diagram plunger the vapors, that is the collar of slide-valve plunger by length , placed in bushing (case) with radial clearance (Fig. 47). From the left side of this band, is located the cavity high (p_1) and with right - the cavity of low (p_2) pressure. We assume that the plunger and the bushing are cylindrical and plunger is placed so that its axle/axis is parallel to the axle/axis of bushing; however, is displaced relative to the latter to value e, as a result $y_2 > y_1$.

Under the taken condition of the truth of plunger and bushing, the cross-sectional areas both the upper and lower part of these clearances will be constants for entire length plunger. Consequently, pressure gradient a for upper, that and for a lower clearance it is possible to approximately accept as constant, in accordance with which the pressure in the upper and lower clearances linearly reduced from the input p_1 to the exit p_2, i.e., a and b will be straight lines.

From the given diagram/curve of forces of pressure, it follows that the radial force of pressure of liquid in the upper clearance, which attempts to displace plunger down, in this case is equal, if we disregard the velocity pressure head of the flow of leakages, to force of pressure of liquid in lower clearance, that attempts to displace plunger upward, i.e., these forces they are balanced.

In connection with this it should be noted that the complete balancing of plunger will occur only with its concentric (coaxial) location in case, in its eccentric position it appears the lateral pressing force, caused by a difference in the velocity heads.

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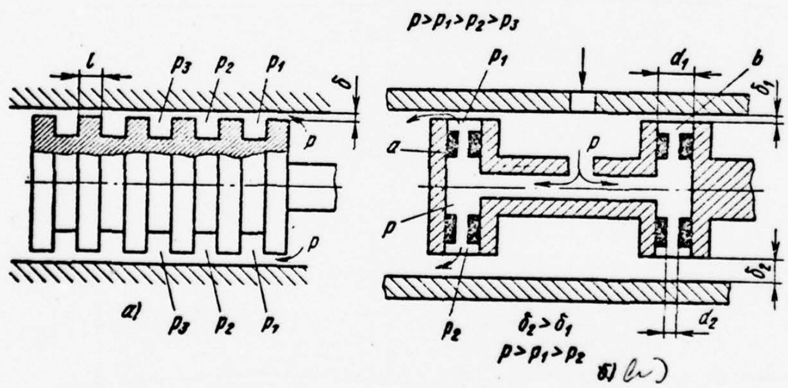

Fig. 48. Diagrams unload slide valve from the influence of the unbalanced radial forces of pressure of liquid.

With the disturbance/breakdown of the parallelism of the generatrices of the slot, caused by the distortion of the cylindricity of the surfaces of bushing and plunger, by the bending of axle/axes or by other reasons, but the also when, on these surfaces, avalanches and other production defects are present, the steadiness of radial forces will be disrupted, and will arise the forces, which approach to displace plunger one side or the other of the surface of bushing. As a result under known conditions, can arise the frictional forces, considerably (hundred times) the exceeding forces, which act into slide valve during the satisfactory production of its parts.

The frictional forces of plunger can be raised to also as a result of the deformations of housing it can occur unevenly, in view of which the form of opening/aperture under plunger can be distorted is concealed by form, that on some sections the clearance will increase, and on others it decreases, as a result the plunger of valve can turn out to be that which was pinched.

The source of an increase in friction are the solid particles of the contaminations of liquid and the tarry-asphalt formation/education. The solid particles, caught into clearance, act

on the plunger of slide valve in radial direction, and also can injure (scratch) effective surfaces.

The force, which requires for the moving of plunger, depends also on the physical properties of liquid and is connected with the cicatrization (obliteration) of slot by the adsorbed on the surfaces of parts molecules, i.e., it depends on the properties of the boundary layer of liquid. during the complete cicatrization of slot, will occur the "union" of the surfaces of bushing and plunger by the fixed layers of these molecules.

In this case in order to move (to move) plunger from place, necessary to make the effort, capable of breaking the layer from these molecules, which connects the surfaces of bushing and plunger.

For a reduction in the frictional force, it is necessary first of all to decrease the unbalanced radial forces of pressure of liquid on plunger. The simplest method of a reduction in the indicated forces is the cutting on the ram area or case of circular rectangular grooves (Fig. 48a). Since the hydraulic friction of groove with its appropriate section is negligibly small in comparison with the

friction of radial slot, in each of grooves, will act identical for any point in vicinity pressure, whereupon $p > p_1 > p_2 > p_3$ ...., as a result for plunger, will act the unbalanced pressure only at length belt/zone (cross connection) between grooves. It is obvious, with infinitely fine/thin bands lack of balance completely it will disappear in any positions of plunger.

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For the discharging of valve from the action of the unbalanced radial forces, is applied also hydrostatic centering of plunger. one of the diagrams of similar centering is represented in Fig. 48b. In the bands of plunger, they make several (4-6) oppositely arranged/located radial opening/apertures (d_1 = 6-8 mm), connected with axial channel with the cavity of the power supply of valve. These opening/apertures are overlapped by the embedded in them plugs a, in which are executed the choke opening/apertures of a small section through which is fed liquid from the feed chamber into chambers b, formed by the embedded plugs and the bushing of valve.

It is not difficult to see that if as a result of the radial displacement of plunger upward the clearance on the upper side

decreases, then on opposite it by the same size/dimension will increase ($\delta_1 < \delta_2$), as a result of which pressure p_1 into the chamber from the side of the reduced clearance as a result of the action of throttle/chokes will be raised, and pressure p_2 in the chamber from the side of increased clearance is lowered ($p_1 > p_2$), and consequently, will appear the unbalanced radial force, which attempts to displace plunger toward the position, coaxial with bushing. When this force overcomes the forces, which attempt to decentralize the plunger, last/latter, after displacing in direction to the axle/axis of bushing, begins to hang on oil "cushion". Since the plunger in this case will lose contact with the surface the forces necessary for future reference its displacement from place and it will be lowered to the value, caused by the fluid friction which hundred times are less than the force, which requires in the plunger vapors without this discharging.

For the reliable self-centering of plunger in the diagram of discharging in question it is necessary that the hydraulic friction of radial slot in the coaxial position of plunger and bushing, otherwise the centering effect of the pressure of liquid in chambers b decreases to zero.

To application/use in practice it can be recommended with the widespread diametral clearances (10-20 μm) five of throttle opening/apertures 0.2-0.3 mm in diameter. With less clearances the diameters of opening/apertures can be decreased to 0.15 mm. In3 for a decrease in the frictional forces are applied valves, the plungers or the cases, which accomplish the forward/progressive or rotary vibration (oscillating) fluctuations of small amplitude (10-100 μm) and of high frequency (>50 Hz). These fluctuations are realized with the aid of different mechanical and electrotechnical means.

Experiment shows that the force, necessary for the moving of plunger from place during oscillating motions, especially with axial, composes small part (3-4o/o) of the forces which are necessary for this in the absence of such motions.

The application/use of reciprocating fluctuations is especially expedient in the valves of the servo systems in which the plunger accomplishes the oscillatory motions of the relatively its mid-position. The amplitude of oscillations must somewhat exceed (on 10-50 μm) the overlap by the collars of the plunger of the windows of power supply. With the oscillations of plunger with this amplitude,

all the driving power packs of hydraulic system will be subjected to frequency alternating loads, caused by the alternating forces of pressure of liquid on the piston of actuating cylinder, thanks to which it descends friction in all these node/units. Oscillation frequency in this case must be such that the piston of actuating cylinder does not react to the momentum/impulse/pulses, caused by the frequency fluctuations of the supply of liquid into cylinder. Similar frequency is the frequency above 50 Hz.

end section.

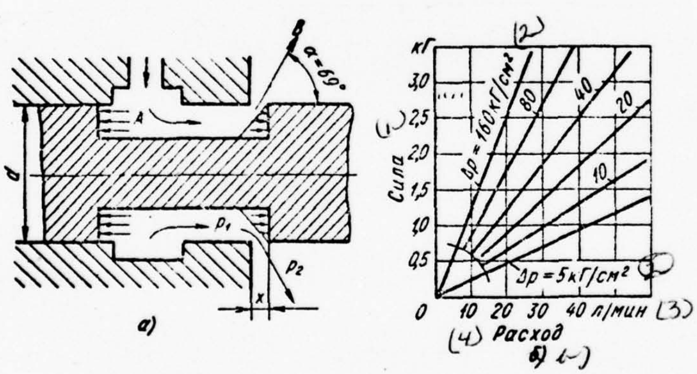

Fig. 49. The design diagram of the action of hydrodynamic (jet/reactive) forces on the plunger of valve (1) and the curve/graph

of the forces, which act in the valve, which does not have the means for discharging (b).

Key: (1). Force. (2). kgf/cm². (3). l/min. (4). Expenditure/consumption.

on the plunger of slide valves, act also the axial forces, produced by the hydrodynamic (jet/reactive) action of fluid flow. The indicated forces act to the side of the coverage of slide valve ports (they attempt to return plunger to free position).

Experiment shows that this force is the function of two variables - the pressure differential of liquid on the plunger of valve and discovery/opening the window of the latter, i.e., it depends on the lost as a result of throttling/choking in valve power. Virtually it is possible to consider that in standard fourway valves to every 1 h.p. of the power, lost in valve as a result of throttling/choking in expenditure slot with a width of x (see Fig. 40), comes the axial force, equal approximately 0.4-0.6 kgf.

From the diagram of the plunger pair, presented in Fig. 49a, it follows that the fluid flow from chamber A into chamber B through slot x of expenditure window, formed by the edges of plunger and bushing, is sloped toward the axle/axis of plunger at an angle α, computed value of which during the establish/installed turbulent flow conditions for a plunger pair with zero clearance and the sharp edges of windows can reach value $\alpha = 69°$.

Reaction force R of fluid flow, that acts at an angle to the axle/axis of plunger in the direction, opposite to the direction of the speed of flow, it is possible to approximately determine from the expression of the momentum of jet

$$R = mu_T,$$

where m and u_T - the mass fluid flow rate, which takes place through the slot of window, and the average rated speed of flow at output/yield from slot.

Taking into consideration that

$$m = Q\rho,$$

equation (37) it is possible to rewrite in the form

$$R = Qu\rho,$$

where Q is volumetric fluid flow rate through the slot of window.

The axial component of this force will be determined from equation

$$F = R\cos\alpha = Qu_T\rho\cos\alpha.$$

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After substituting

$$Q=\mu f\sqrt{\frac{2\Delta p}{\rho}}=\mu\varpi x\sqrt{\frac{2\Delta p}{\rho}},$$

and also the calculated jet velocity

$$u_T=\sqrt{\frac{2}{\rho}\Delta p},$$

into the preceding/previous expression, we will obtain approximately expression for the force of valve with zero clearance and the sharp edges of expenditure windows

$$F = 2\mu f\, \Delta p \cos\alpha = 2\mu w x\, \Delta p \cos\alpha,$$

where $f = wx$ - the area of the slot of window; w is length of slot on circular arc (for a plunger with annular groove $w = \pi d$); $\Delta p = p_1 - p_2$ is the pressure differential in the slot of the window between chambers A and B; x is discovery/opening window.

Since angle α is positive, force F attempts to displace plunger to the side of a decrease in the expenditure window, in other words, the hydrodynamic action of the liquid, passing through the slot, formed by the edges of plunger and windows of bushing, analogous with spring effect, which attempts to return the displaced plunger to free position. The approximate value of this force for a standard valve in the function of expenditure/consumption for different pressure differentials $\Delta p = p_1 - p_2$ is given in Fig. 49b.

Assuming that flow forces in fourway valve (see Fig. 40) with the symmetrical location of the bands of plunger relative to the windows of bushing it will be with the equal expenditure/consumptions of forcing and gutter doubled (this force acts both in worker and in the overflow chamber of valve), the total thrust under this condition will be also doubled:

$$F_c = 2F = 2Qu\rho\cos\alpha = 2Q\sqrt{2\rho\,\Delta p}\cos\alpha$$

or

$$F_c = 2Q\sqrt{\rho\Delta_c}\cos\alpha,$$

where $\Delta p_c = 2\Delta p$ it will be the total (for two chambers of valve) pressure differential.

Two-stage slide-valve distributors.

If necessary for a considerable reduction in the force, which requires for the displacement/movement of the plunger of valve it is simultaneous for providing the required fluid flow rate, apply the two-stage (two-stage) distribution valves which were called the name servo slide In these distributors between the setting device and the plunger of the basic distribution valve, is establish/installed the amplifying circuit, as which serves the intermediate (auxiliary) valve.

The schematic diagram of a similar valve is shown in Fig. 50a.

Basic distribution valve 3, which feeds performing hydraulic engine 1, is controlled with the aid of auxiliary valve- sensor 2 of small section. For the drive of this auxiliary valve of two-stage distributor, are commonly used the electromagnets or the electric motors. Since the plunger of auxiliary valve usually has small size/dimensions (diameter of approximately 3-4 mm), for its drive can be applied low-power electromagnet.

one of the diagrams of two-position distributor with electromagnetic control is depicted on Fig. 50b. The plunger of 6 basic valve with connected electromagnet 2 is held on end left position by the action of the forces of spring 3 and by the pressure of the liquid, which enters from working main line chamber c through channel b. In this position of valve, the liquid from working main line 5, connected with pump, enters channel e, connected with actuating cylinder. With connected electromagnet 2, liquid from main line 5 will be fed through the annular groove of auxiliary valve 1 and channel a into the left cavity f of plunger 6.

Fig. 50. Diagrams of two-stage distribution valves.

Fig. 51. Valve with flat/plane distributive cell/element.

Key: (1). From pump. (2). To engine. (3). In tank. (4). From engine.

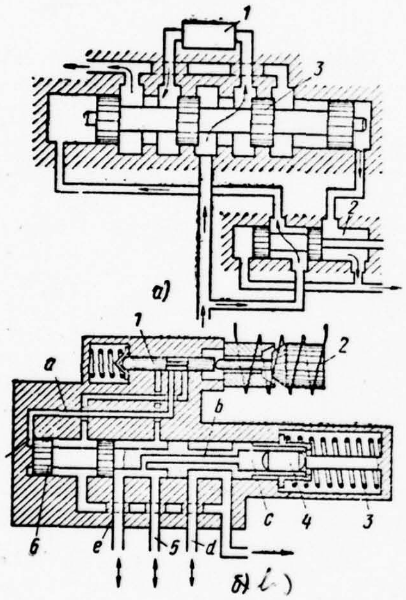

Fig. 50

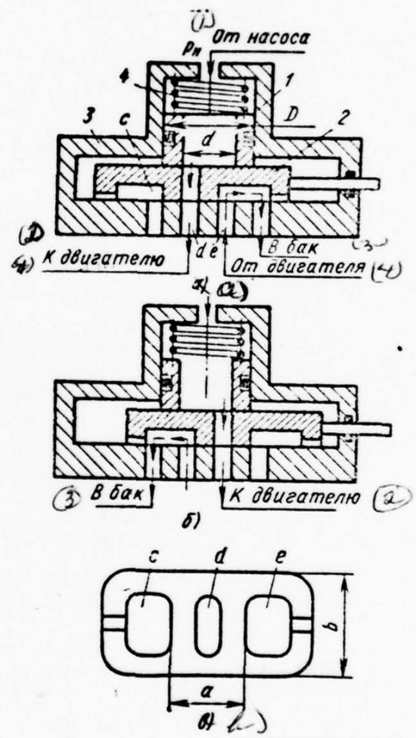

Fig. 51

Since the effective area of plunger 6 is greater than the area of plunger 4 of auxiliary valve, plunger 6 will move to the right. In this case, working main line 5 will be connected with channel d, connected with the second cavity of actuating cylinder.

Slide valves. The basic difficulty during the production of the sleeve valves of high accuracy/precision is connected with the complexity of working and quality control of the internal effective surface of bushing. In view of this of interest is the construction in which is provided for the admission to this surface.

Figure 51a shows the diagram of distributor with flat/plane distributive cell/element (valve), which satisfies the latter requirements.

Slide valve 2 slips on the flat/plane basis/base (mirror) of housing 3, being pressed against it through bushing 1 by spring 4 and by the force of the pressure of liquid.

The liquid through the central through window d enters depending

on the position of valve the left (Fig. 51a) either right (Fig. 51b) window of the power supply of hydraulic engine it is driven out from the nonoperative cavities through the appropriate drained windows e or c (see Fig. 51c).

Force P_1, with which valve 2 is pressed against the mirror of housing, will be determined (not allowing for the force of spring) by a difference in the forces of pressure of the aaaa of working fluid on the washed by it area of the bushing of 1 and valve itself 2, which press the latter to the mirror of housing, and forces P_2 the mean pressure of the aaaaa of liquid in the flat/plane clearance, formed by valve and this mirror, that wrings out valve from housing. Disregarding the effect of the area of the through window d, which enters in the expressions both of the pressing and wringing out forces, and assuming that pressure distribution in flat/plane clearance bears the linear character with which

$$p_{cp}=\frac{p_H}{2},$$

and also accepting, that this pressure acts on area f = ab (Fig.

51c), the equilibrium condition of the acting on valve forces can be presented expression

$$P_1 = P_2; \quad \frac{\pi D^2}{4} p_{\kappa} = ab \frac{p_{\kappa}}{2}$$

or

$$D = \sqrt{\frac{2ab}{\pi}}.$$

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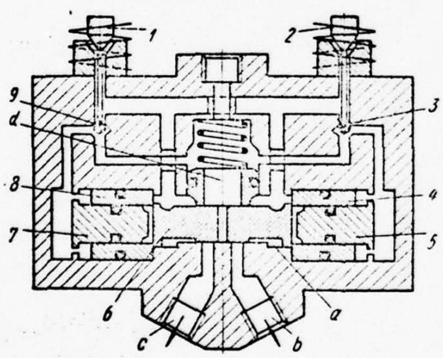


Fig. 52. Slide valve with servo effect.

In order that would not occur the "expansion/disclosures" of distributor (breakaway of valve from housing), must be observed condition $P_1 > P_2$.

Practice shows that the expansion/disclosures is not observed under condition

$$\frac{\pi D^2}{4} p_{\kappa} = ab \frac{p_{\kappa}}{2} 1{,}1.$$

In this case the wringing out force will be to 10o/o less pressing force to which is added also the force of spring, selected so that would be provided for close contact at zero and low pressure in system taking into account friction of the ferrules of bushing 1.

The advantage of the distributors in question with slide valve are the less rigid, than to the valves examined above with the

plunger of requirement with respect to purity/finish and the accuracy/precision of working, and also the possibility of obtaining high (virtually absolute) airtightness.

An increase in the hydraulic slipes with an increase in the temperature occurs in flat/plane distributors less intensely (5-6 times), than in sleeve valves, which is explained by the automaticity of compensation for play between the being moved parts. Since this clearance is determined by the thickness of carrying oil film, hinder/hampered also penetration in it in view of its smallness of the solid particles of the contaminator, thanks to which distributors differ in terms of sufficiently high service life. Furthermore, in view of the free orientation of movable cell/element relative to motionless there is no danger of its wedging in the case of the incidence/impingement into the clearance of solid contaminations, and also as a result of the temperature expansion of materials.

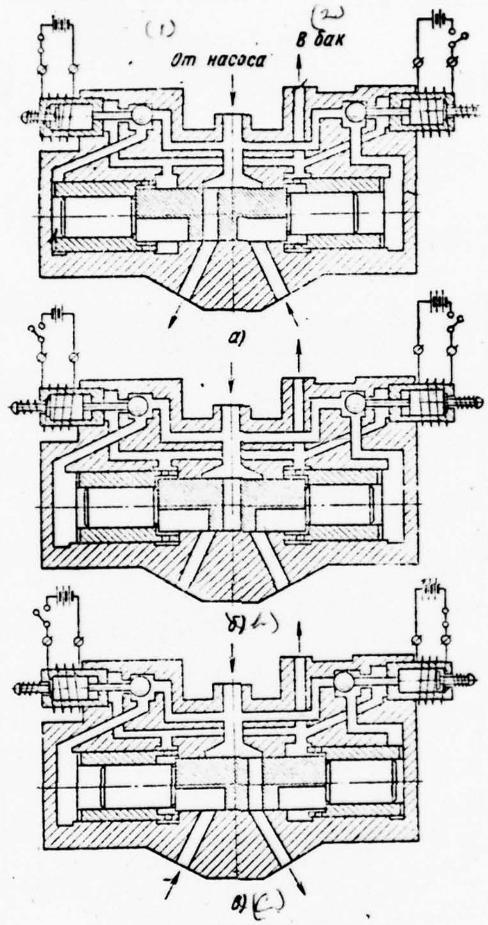
The planes of distributor manufacture both with direct drive of distributive cell/element (Fig. 51) and with servo effect. The diagram of the last/latter distributor is depicted on Fig. 52. It is the fourway three-position slide valve which in free position disconnects the working lines, connected with actuating cylinder,

from the source of pressure and discharge lead.

Ball bearing valve- sensors 9 and 3 are controlled by the successive switching on of electromagnets 1 and 2. With the disconnection (de-energizing) of electromagnets, valves 9 and 3 are pressed against the upper saddles, and liquid under pressure it passes through these valves, exerting identical action on internal 5 and 7 and external 4 and 8 circular pistons. External pistons 8 and 4 are force against detents at the internal ends of the cylinders, in view of which slide valve 6 is centered (it is establish/installed in free position), blocking the working exit windows c and b. Upon the switching on of electromagnet, 1 corresponding valve 9 is wrung out at downhand position, disconnecting pressure line from pistons 8 and 7 and connecting them with the discharge lead, which goes to reservoir, so that piston pressure decreases.

Fig. 53. Diagrams of the action of slide valve with servo effect.

Key: (1). From pump. (2). In tank.



In this case, the pressure of the liquid, which acts on internalizations piston 5, will transfer valve 6 at the position by which the liquid under pressure will pass through the central window d and further past the working window of valve 3 to cylinder.

The liquid, displaced from the noroperative cavity of cylinder, enters the distributor through second channel b which in this case is connected through chamber a with drainage line. After the disconnection of electromagnet, 1 valve 9 returns to its upper saddle, as a result of which the equilibrium of forces of pressure is reduced, and valve returns to central position.

Upon the inclusion of electromagnet 2, system acts in opposite direction. in this case the liquid is directed to toward channel b, and window c is connected with drainage line.

Figure 53 shows the positions of the working cell/elements of distributor. Unlike previously examined diagram of distributor (see Fig. 51) this distributor with switched on electromagnets does not block the cavity of actuating cylinder, but it connects them with tank. In the position of distributor, represented in Fig. 53a, is

included left electromagnet also in Fig. 53c - the right electromagnet; in the position, presented in Fig. 53b, are turned off both electromagnets.

Crane distributors. Plug tap/cranes find a use as distributors at small expenditure/consumption and pressure.

In crane distributors a working cell/element (plug) of the cylindrical or conical (Fig. 54a) type accomplishes rotary relative to its axis of movement. Cock plug must be balanced from the static forces of pressure of liquid, since otherwise it will be force against one side, in consequence of which can be developed large frictional forces. Balancings in the tap/crane, diagram which is represented in Fig. 54b, achieve by the diametrically opposite effect of pressure of liquid on plug. The supply of liquid is realized through channels a and the power supply of hydraulic engines - through channels b, the tank connect with channels c.

For a decrease in friction, the slewing cranes of servo systems frequently are centered on antifriction bearings, for example needle-shaped (Fig. 54c).

In tap/cranes with cylindrical plug, the sealing contact is provided by grinding, while in miter plug valves - with the aid of the spring (see Fig. 54a) whose force must exceed the counteraction of the pressure of the liquid, which attempts to eject tap/crane of seat/socket.

Fig. 54. Diagrams of crane distributors.

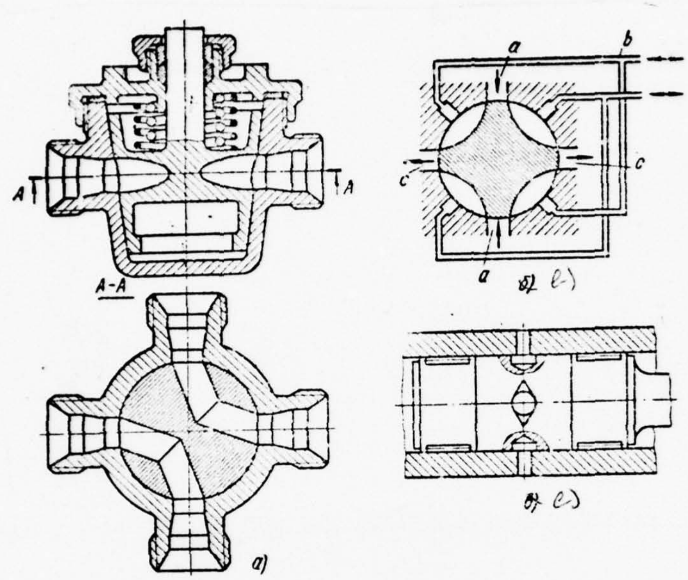
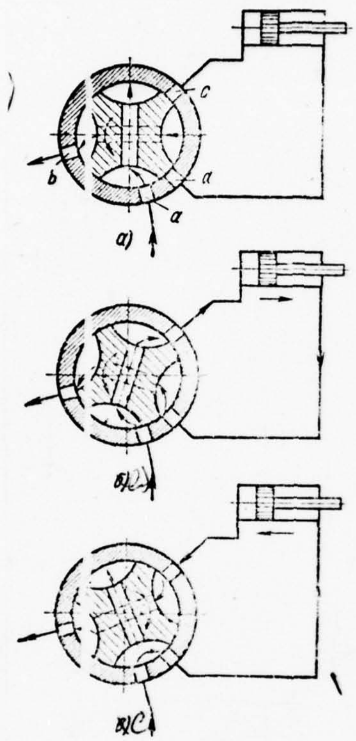


Fig. 55. Circuit diagrams into the hydraulic system of plug tap/crane.



Since the spring is designed for the maximum operating pressure, at low and zero pressures for the turning of tap/crane are required considerable forces, also, in particular, if the latter is designed for high pressures. In view of this such tap/cranes are applied in pressure <100 kg/cm².

Figure 55a-c shows the circuit diagram of the widespread crane distributor into the control system of actuating cylinder. Pump connect with channel a tank with channel b; channels c and d - with hydraulic engine.

Cock plug has two perpendicular, but not intersecting opening/apertures. It can occupy two and more than angular positions. Similar tap/cranes are applied as independent distributor, and also as tap/crane-pilot.

Slewing crane with flat/plane distributive cell/element. Are applied also distributors (tap/cranes) with rotary flat/plane distributive cell/element (valve).

The diagram of the simplest tap/crane of this type is represented in Fig. 56a. Flat/plane rotary cell/element 1 has two sickle-shaped groove/slots (window) c and g. With the aid of the anechoic groove/slot g, channel f, which drives to Baku, consecutively is connected during the rotation of tap/crane through 90° by channels e and d, which drive to hydraulic engine; in turn, these channels with the aid of groove/slot c and open-end hole b in the rotary cell/element of 1 successively connected with the input chanel h, connected with pump.

Fig. 56. Tap/cranes with flat/plane rotary cell/element.

Key: (1). In tank.

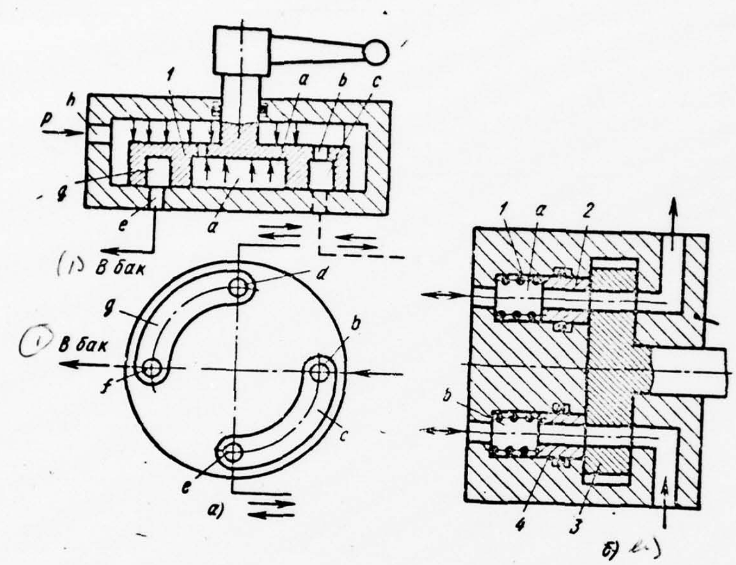
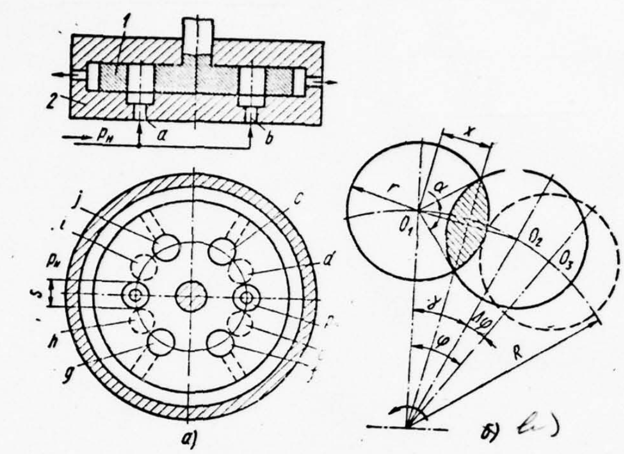


Fig. 57. Flat/plane rotary valve (a) and its design diagram (b).



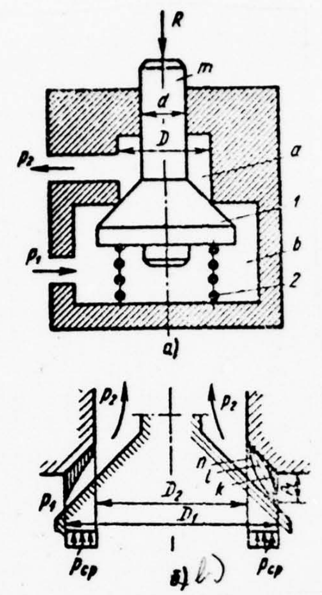
For the discharging of rotary cell/element from the forces of the operating pressure of liquid in the latter, is made chamber a, connected with the cavity of operating pressure. The force of pressure of liquid in this chamber resists the force, which presses rotary cell/element 1 to the basis/base of tap/crane.

For the discharging of movable cell/element, are applied also the tap/cranes with cylindrical bushings (Fig. 56b), the action of the mechanism of discharging of which is based on the same principle, as the action of the examined slide valves of forward motion (see also Fig. 51). The airtightness is provided by bushings 2 and 4, by adjusted springs 1 and by the pressure of liquid in channels a and b, connected with hydraulic engine, to the flat surface of rotary valve 3.

In slave/servo type, hydraulic drive are applied similar type precision valves with rotary flat/plane cell/element and zero overlap. The diagram of a similar valve is shown in Fig. 57a. Liquid will be fed from main pressure line to opening/apertures a and b of housing 2. In the free position of rotary slide valve, 1 these opening/apertures overlap with the cross connections between the expenditure windows i, h, d and e of this valve, connected with the

cavities of hydraulic engine.

Fig. 58. Cell/elements of valve distributor.



The width s of this cross connection determines the size/dimension of the overlap: with $s = d$, where d - the diameter of opening/aperture a, overlap is equal to zero. In tank the liquid is abstract/removed through opening/apertures k, c, f and g.

In the valve in question are applied the paired windows for an increase in the area of channels and obtaining the even distribution of load on rotary valve.

Under the condition of the equality of the diameters of opening/apertures a and b of power supply (distributive windows) and during zero overlap ($s = d$) the angle between centers O_1 and O_3 distributive windows in bushing and in valve (Fig. 57b) at the torque/moment of the beginning of their coincidence (contact) is equal to φ. During the rotation of valve in the direction of the coincidence (in the direction of arrow/pointer) of windows to certain angle $\Delta\varphi$ angle γ between the centers of these opening/apertures in their new position (between centers O_1 and O_2)

$$\gamma = \varphi - \Delta\varphi.$$

The distance x, the equal approximately half of distance between centers O_1 and O_2, the angle α are determined from geometric relationships

$$x = R\sin\frac{\gamma}{2} = R\sin\left[\arcsin\frac{r}{R} - \frac{\Delta\varphi}{R}\right];$$

or $$\cos\frac{\alpha}{2} = \frac{x}{r} \text{ или } \alpha = 2\arccos\frac{x}{r}.$$

The Area of the segment which is formed by the circular arc of window with a radius of r, equal to the half of the area of passage slot, formed during rotation through angle $\Delta\phi$ (during the combination of round passage opening with rectangular),

$$f=\frac{r^2}{2}\left(\frac{\pi\alpha}{180}-\sin\alpha\right).$$

Complete area of discovery/opening slide-valve slot during the combination of two round opening/apertures of equal diameter

$$f_{\text{пол}}=2f=2r^2\left(\frac{\pi\alpha}{180}-\sin\alpha\right).$$

Valve distributors. In the hydraulic systems of some machines, are applied also the valve distributors which are simple in production and are reliable, and also they can ensure high

(practically complete) airtightness. The diagram of valve is shown in Fig. 58a.

The gates of valves are put into action by manual, mechanical and electrotechnical devices. From manual devices are most common the valves with the rocker arm whose diagram for the power supply of one cavity of the hydraulic engine of reciprocal action is given in Fig. 59a. In the neutral (average) position of rocker arm 1, both valves 2 and 3 are located in its seat/sockets; in this position of valves channel b of hydraulic engine is disconnected both from the channel a, connected with pump, and from channel c, connected with tank. During the rotation of lever 1 to the right with hydraulic engine will be connected channel a of pump, during rotation to the left it will be connected channel c of tank.

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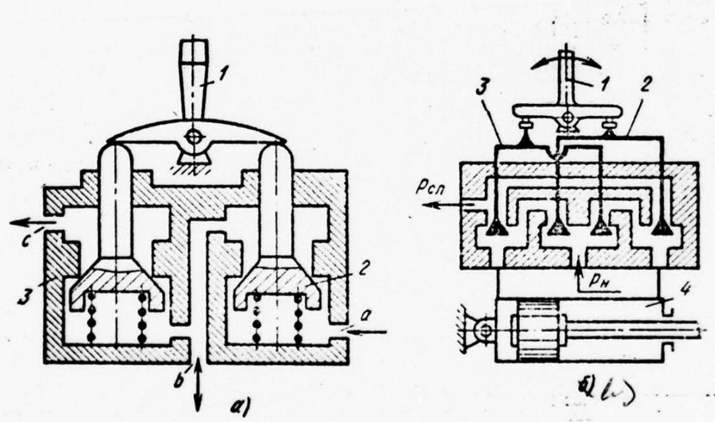

Fig. 59. Diagrams of valve distributor.

The diagram of fourway valve distributor is represented in Fig. 59b. During the rotation of crank 1, is moved one pair or the other of valves 2 or 3, providing the supply (diversion/tap) of liquid to the appropriate cavity of actuating cylinder 4.

$$R = p_1F_1 + P_{np} + S_n - p_2(F_2 - f),$$

where F_1 and F_2 are surface areas of the contact of gate with seat/socket according to diameters D_1 and D_2 (see Fig. 58b).

If one assumes that the force of the tightening of the spring of P_{np} and pressure p_1 liquid after gate is detached away from saddle, they will not change and pressure in the formed slot it will decrease on linear dependence from p_1 to p_2 (see the curve of k in Fig. 58b), then force R_1, necessary for the further displacement/movement of gate after it is detached away from saddle, it will compose

$$R_1 = p_1F_1 + P_{np}S_\partial - p_2(F_2 - f) - p_{cp}(F_1 - F_2),$$

where the S_∂ - the frictional force of motion; p_{cp} = $p_1 + p_2/2$ - mean pressure in slot.

Consequently, force R is greater than force of R_1 on

$$\frac{p_1+p_2}{2}(F_1-F_2),$$

and also to a difference in the frictional forces of the rest of S_n and movement of S_∂, in accordance with which after the breakaway of gate from seat/socket the force, necessary for its further displacement/movement, will be lowered.

After discovery/opening gate, the pressure in cavity a will be raised to $p_1 = p_2 + \Delta p$, where Δp it will be raised a pressure increment in cavity 2 after discovery/opening gate. In accordance with this, will be raised also the mean pressure in conical slot

$$p_{cp} = \frac{p_1 + p_2 + \Delta p}{2}.$$

Thus,

$$R_1 = p_1F_1 + P_{np} + S_{\partial} - (p_2 + \Delta p)(F_2 - f) - \left(\frac{p_1 + p_2 + \Delta p}{2}\right)(F_1 - F_2).$$

It is obvious that the pressure distrubution law along the length of the slot of valve distributor, which is found, can be also exponential, l of the distribution curve of pressure can be both convex and concave (see Fig. 58b), that it will lead to a change in

the given lining/calculations.

Are common also the valves with cam drive, diagram of one of which is depicted on Fig. 60a. On cylinder 3, are located four cam/catch/jaw 2, in an appropriate manner oriented one relative to another.

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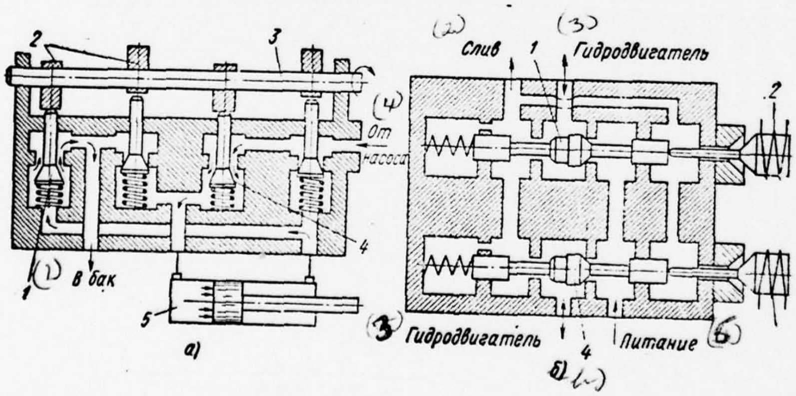


Fig. 60. Valve distributor with cam (a) and electromagnetic (b) drive.

Key (1) In tank (2) Gutter: (3) Hydraulic engine; (4) From pump; 5) Power supply.

During the rotation of cylinder, the cam/catch/jaws affect the stock/rods of corresponding conical gate 1, providing the supply of liquid in the cavity of actuating cylinder (hydraulic engine) 5 and its diversion/tap. In the position, presented in Fig. 60a, the liquid from the channel, connected with pump, comes through discovery/openings (embedded) gate 4 into the left cavity of actuating cylinder 5 and is driven out into tank from the right cavity of the cylinder through the valve. The others two gate are located with its saddles. during the rotation of cylinder, enter into action these gates, providing the supply of liquid into the right cavity of cylinder 5 and its diversion/tap from left cavity.

Figure 60b depicts the diagram of the three-position valve action of direct distributor with two valves 1 and 4, controlled electromagnets 2 and 3. With switched off electromagnets both valves are pressed by springs to its saddles. In this case, the main line of forcing is overlapped, and the cavities of hydraulic engine (user) are connected with gutter.

Upon the switching on of electromagnet 2, valve 1, compressing spring, will move into end left position and will snuggle up itself to left saddle. In this position one of the cavities of users will be

connected with the main line of forcing. With the connected electromagnet 3 and switched off electromagnets 2, will actuate/operate valve 4, after connecting the second cavity of user with the main line of forcing.

Acting forces. Force R (see Fig. 58a), which must be applied to shank m of the gate of 1 valve with the sharp sealing edge for an elevation or its retention in the built up position (without taking into account the reaction forces of fluid flow) and assuming that the pressure of medium on the external end/face of shank m does not act, it is possible to calculate according to expression

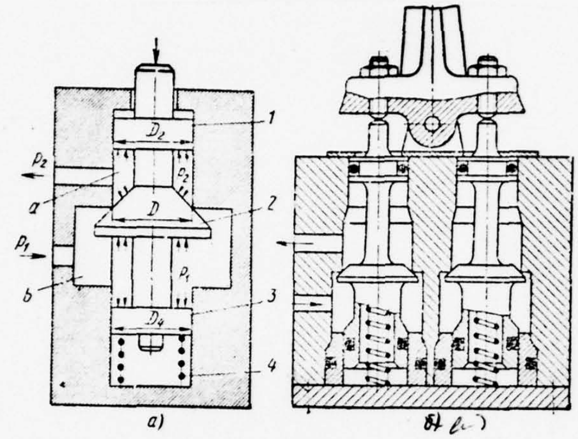
$$R = p_1F - p_2(F - f) + P_{np} + S_n,$$

where p_1 and p_2 acts pressure respectively in cavity a of the supply of liquid to distributor and in cavity b of the diversion/tap of liquid into the system of users; $F = \pi D^2/4$ - the surface area of the contact of the conical part of gate 1 with saddle; $f = \pi/4\ d^2$ is an area of shank m of block; P_{np} - the force of the pretightening of spring 2; S_n - the frictional force of rest.

Virtually the contact of the gate of distributor occurs not on sharp edge, but on the cone of the saddle (see Fig. 58c), in view of

which the forces, which act on gate, will depend under these conditions on the width of its contact surface with saddle.

Fig. 61. Diagrams of the discharging of valves from the axial forces of pressure of liquid on gate.



If in the conical slot, formed by the surface of gate with the surface of seat/socket, pressure is absent (contact occurs on outward flange), then to shank m of gate for its breakaway from saddle it is necessary to apply force.

Methods of the discharging of valve. A deficiency/lack in the examined valve distributors are the large forces which are required for the overcoming of the pressure of liquid on the surface of the gate of valve.

For the discharging of the gate of valve from the force of pressure of liquid, are applied the different structural/design means, one of which is the balancing of this force the force, oppositely directed.

Figure 61a depicts the design diagram of valve with the balancing of forces of pressure, which act both at the inlet and at the output/yield of valve. Valve 2 in its lower part is equipped that which balance piston 3. If diameter D_4 this piston is equal to diameter D of valve seat, then the latter is unloaded from static forces of pressure p_1 liquid in chamber b.

For the discharging of valve from pressure p_2 liquid in chamber a is applied second piston 1 by diameter D_2.

From the examined diagram it is evident that under condition $D = D_4 = D_2$ the valve is completely statically balanced from the forces of pressure of liquid. To saddle it in this case is pressed only by the force of spring 4. In the case $D_4 < D$ to the force of spring, will be added the force of pressure of liquid

$$P = p_1 \frac{\pi (D^2 - D_4^2)}{4},$$

pressing valve to saddle.

In order to compensate for after the breakaway of valve from saddle (with $p_2 > 0$) the force of spring, is accepted $D_2 > D$. The construction of valve with similar discharging is given in Fig. 61b.

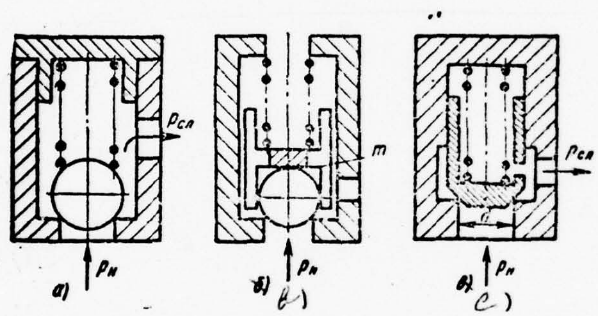


Fig. 62. Diagram of safety valves.

The valve is in the general case the equipment/device, intended for a control of fluid flow by automatic changing the working window under the effect through it of the working fluid taking place. In hydraulic drive the valves are utilized in essence as the pressure regulators and fluid flow rate. Pressure regulators are subdivided into protective (overflow) and reduction valves, flow regulators are subdivided into stabilizers and the limiters of expenditure/consumption, and also the divider/denominators of flow and check valves.

Safety valves restrict a pressure increase in system over that which was assigned by means of the periodic and single diversion/tap (letting out) of liquid into tank. Overflow valves are intended for maintaining pressure in system by means of the continuous letting out of liquid as, for example, with the choke of regulation of expenditure/consumption (rate of hydraulic engine).

Are distinguished action of direct valves and two-stage valves (with servo effect). In action, of direct valves the size/dimensions of working window change as a result of immediate effect on the locking-controlling organ/control (gate) of the passing through it flow of working fluid. In valves with servo effect, the

size/dimensions of working window (windows) change as a result of the effect of fluid flow to the locking-controlling organ/control through intermediate means. Is applied also name "valve of the pressure" hearth by which is understood the controlling hydraulic apparatus, intended for a control of the pressure of liquid, and also the "pressure valve", intended for the limitation of pressure in the supplying hydraulic lines, whereupon depending on the performed function these valves they can be called protective and support valves.

Action of direct valves.

The operating principle of such valves, used in the hydraulic systems of machines, is based on balancing by the external force (spring, electromagnet, load, etc.) of the force of the pressure of liquid, which acts on the gate of valve (ball/sphere, the plunger with conical landing place, etc.), which under the action of this force tightly (hermetically sealed) overlaps passage channel (Fig. 62).

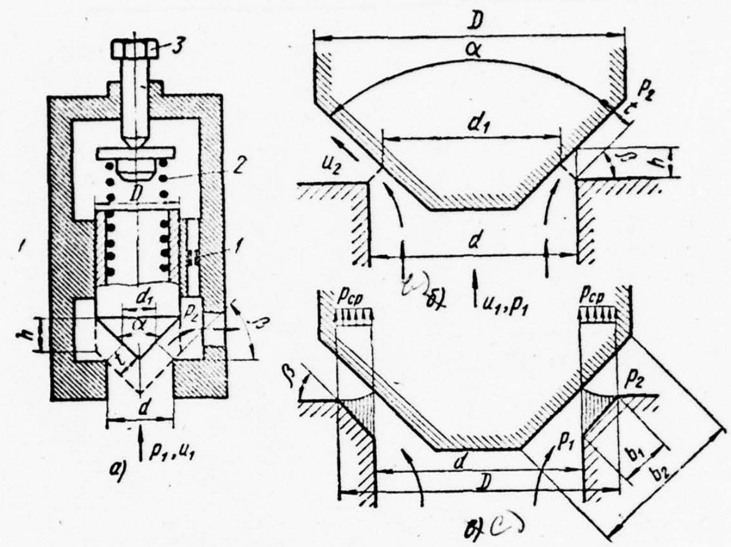
After the force of pressure of liquid; acting on the gate of

valve, will overcome this reactive external force, gate, after displacing from its saddle, will discover passage for a liquid into drainage line (into tank). With decompression at the inlet into valve lower than the value, which corresponds to the reactive external force, the gate again will overlap the passage of liquid into tank. In accordance with this, the safety valve is the throttling equipment/device (organ/control) with variable flow passage cross-sectional area.

Simplest of the safety valves is ball bearing with the constant (Fig. 62a) or adjustable tightening of spring. However, these valves are used only during relatively small pressures and during short-term action, since in continuous operation ball/sphere as a result of vibration unevenly develops (it divide/marks off) the seat/socket of valve. For a decrease in this nonuniformity of the consumption/production/generation of saddle ball/sphere, in particular in the valves of the systems high pressure, they equip with guide m (Fig. 62b), with the aid of which is provided its displacement/movement only along axle/axis.

This same type includes the valve with the conical gate whose diagrams are depicted on Fig. 62c and 63a.

Fig. 63. The design diagrams of safety valves with conical gate.



The necessary conditions of providing an airtightness of the last/latter valve is the observance of a strict axial alignment of cylindrical and conical surfaces of gate, and also the axial alignment of the directing cylinder of the valve body and conical seat/socket.

The control of the precompression of spring 2 (Fig. 63a) is realized with the aid of bolt 3. For oscillation damping is provided throttle/choke 1.

Characteristics of valve. The quality of safety valve is estimated at its static and dynamic characteristics.

Static characteristic expresses the dependence between the input and output values in steady-state mode at the different, but constant loads. For valves such characteristics usually express the dependence of pressure p and of displacement/movement h of gate in the function of expenditure/consumption of Q ($p = f(Q)$ and $h = f(Q)$).

Dynamic characteristic describes the transient process, which

proceeds in valve in the period of the displacement/movement of gate and change in the load, expenditure/consumption etc. The last/latter characteristics include also the frequency characteristics, taken in the mode of forced oscillations.

The calculation of valve for a work in static behavior is reduced to the determination of the area of the working window, necessary for the passage through it the required expenditure/consumption Q of liquid with the assigned pressure differential Δp. Fluid flow rate Q and the pressure differential Δp are connected by equation (20), in which enters the variable area f of working window, that depends on height/altitude h of valve lift, and also the variable coefficient of expenditure/consumption μ.

Figure 64 depicts the experimental curve/graph of the dependence of the coefficient of expenditure/consumption μ valve with the angle of taper of gate of apex/vertex, equal to $\alpha = 90°$, on Re of different pressure differentials $\Delta p = p_1 = p_2$. Re was calculated from expression

$$\mathrm{Re} = \frac{4ur_z}{\nu} = \frac{4Qf}{2\nu f\pi d} = \frac{2Q}{\pi d\nu},$$

where the $r_2 = f/\sigma$ - a hydraulic radius; here $\sigma = 2\pi d$ is wetted perimeter; d - the mean diameter of the section of slot, formed by the gate and valve seat.

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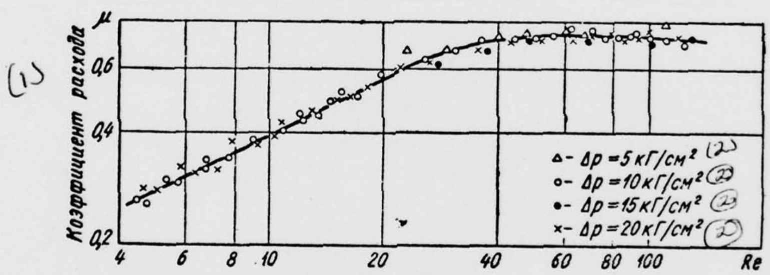

Fig. 64. Dependence of the valve flow coefficient μ on Re.

Key: (1). Coefficient of expenditure/consumption. (2). kgf/cm².

Experiments are carried out on the oil of AMG-10 at of 30 degrees and 50°C.

In this curve sufficiently distinctly observed two section Re < 40 and Re > 40. For the first section (Re < 40) the coefficient of expenditure/consumption can be calculated according to expression

$$\mu = 0{,}126\mathrm{Re}.$$

For the second section (Re > 40), which is basic (predominating) for valves, the coefficient of expenditure/consumption in practice does not depend on Re and can be accepted for this valve μ = const = 0.75.

In a series of recommendations, the of the coefficient of Re $\geqslant$ 40 is taken as μ = to const = 0.8.

Since the section of the slot between gate and valve seat during

lift changes, during the calculation they take the average value of its diameter. Specifically, for a conical gate the mean diameter of slot during its lift is determined approximately (see Fig. 63a and b):

$$d_{cp} = \frac{d + d_1}{2}.$$

In accordance with this the current area of the passage slot of valve with conical gate and with the sharp edge of saddle

$$f = \pi d_{cp} t = \pi t \left(\frac{d + d_1}{2}\right),$$

where t is a size/dimension of slot in section, perpendicular to flow direction; d is a diameter of the channel of valve (sharp edges of saddle); d_1 - the diameter of the cross section of the cone of the

gate of valve in the built up position.

From the design diagram of valve (see Fig. 63a and b) it follows that

$$d_1 = d - h \sin \alpha \text{ и } t = h \sin \frac{\alpha}{2},$$

in accordance with which of

$$f = \pi d h \sin \frac{\alpha}{2} \left(1 - \frac{h}{2d} \sin \alpha\right),$$

where h is a climbing range of the gate of valve along its axle/axis; α - cone-apex angle of the gate.

Since h considerably less than d, by the second term of difference possible, in particular during small lifts

(discovery/openings), to disregard, as a result we will obtain the simplified expression

$$f = \pi d h \sin\frac{\alpha}{2}.$$

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In accordance with this expression for a expenditure/consumption through the valve

$$Q = \mu h d \pi \sin\frac{\alpha}{2}\sqrt{\frac{2\Delta p}{\rho}} = Kh\sqrt{\Delta p}, \qquad (38)$$

where the $K = \mu \pi d \sin\frac{\alpha}{2}\sqrt{\frac{2}{\rho}}$ - the conductivity of valve.

Using expression (38), we find the climbing range of the valve in question with the sharp edge of the saddle:

$$h = \frac{Q}{\mu\pi d \sin \alpha/2} \sqrt{\frac{\rho}{2\Delta p}}.$$

Climbing range in valves with α = 60-90° usually is selected equal by h = (0.2-0.3) d. To avoid wedging, the angle α must be $\alpha \geq$ 60°.

For the used in hydraulic drive protective high-pressure valves and low expenditure/consumptions the lift of gate usually

$$h = (0,1 \div 0,5)\, d,$$

where d is a diameter of passage opening in valve seat.

For large expenditure/consumptions and low pressures, are applied the valves with the lift of gate h = (0.25-0.35) d.

In view of the fact that the coefficient of expenditure/consumption μ for poppet valves with the sufficiently sharp edge of seat/socket is retained virtually constant over a wide range of lifts h of gate (see Fig. 64), expression (38) it is possible to present for specific conditions in the form

$$Q = B\sqrt{\Delta p},$$

where the $B = Kh = \mu h d\pi \sin\frac{\alpha}{2}\sqrt{2/\rho}$ - constant for these conditions term.

The preceding/previous expression for Q shows that under otherwise equal conditions the expenditure/consumption is proportional to square root z of jump/drop $\Delta p = p_1 = p_2$ pressure.

The rate of liquid in the feeder (in the opening/aperture of seat/socket) of safety valve during calculations usually is selected

to 15 m/s, but sometimes in high-pressure valves, they select to 30 m/s and above.

Forces, which act on the gate of valve. On the gate of valve, act the forces of hydrostatic pressure and friction, force of the hydrodynamic effect of flow, the lateral force, caused by the dissymmetry of the distribution of the pressure of liquid in radial clearance and by the misalignment of axes of gate and opening/aperture, and also the force of the lateral pressure, produced by the asymmetric action of the force of spring.

During the first stage of the calculations of valve, are considered only the forces of hydrostatic effect.

The pressure differential Δp, which corresponds to the beginning of discovery/opening (or to the end of the coverage) the gate of safety valve, for a valve with the sharp edges of landing saddle (see Fig. 63a), i.e., the equilibrium condition of the static forces, which act on the gate of valve, it is possible to determine (not allowing for the frictional forces and weight of gate) by equation

$$P_0 = \Delta p f_0; \quad \Delta p = \frac{P_0}{f_0} = \frac{Ch_0}{f_0}, \tag{39}$$

where $\Delta p = p_1 - p_2$ - the pressure differential on the gate of valve; here p_1 and p_2 are a pressure at the inlet into valve and in its drainage chamber; $f_0 = \pi d^2{}_0/4$ - the projection of the surface of the gate of valve, washed by liquid under pressure, on the plane, perpendicular to its axle/axis (sectional area of gate on line of contact of its with the edges of saddle); here $d_0 = d$ is a diameter of the circle of this projection (for valves with the sharp edge of saddle this diameter is the diameter d of the opening/aperture of saddle); $P_0 = Ch_0$ - the force of the precompression of spring; C and h_0 - stiffness coefficient and the precompression of spring.

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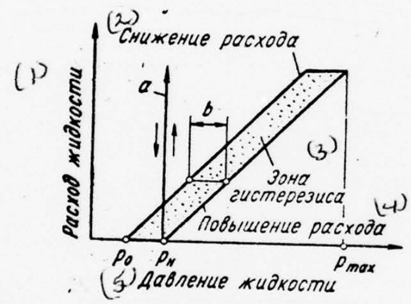

Fig. 65. Dependence of pressure at the inlet in valves on fluid flow rate.

Key: (1). Fluid flow rate. (2). Reduction in the expenditure/consumption. (3). Zone of hysteresis. (4). Increase in the expenditure/consumption. (5). Pressure of liquid.

under this condition the pressure differential Δp, which corresponds to the beginning of valve opening with sharp edge, is equal the pressure differential in the moment of the closing of valve.

After the valve is detached away from its seat/socket, the pressure differential can substantially change. The latter in essence is caused by the emergence of flow forces, and also fact that with an increase in the expenditure/consumption through the valve the gate of the latter must be built up on the appropriate height/altitude, in connection with which the compression of spring and, consequently, also the pressure of liquid, which requires for the retention of the gate of valve in the built up position, they will be raised.

Figure 65 gives the dependence of pressure on the expenditure of single-stage valve during discovery/opening gate (during an increase in the expenditure) and with coverage (with a decrease in the expenditure). Curve/graph shows that the pressure of $p_{н}$, which corresponds to the beginning of valve opening during a pressure increase, is more pressure p_0 at the end of the closing of valve during a decompression. The difference in these pressures with equal expenditures determines hysteresis protective the valve, by which in

the general case is understood the difference between pressures with opening and of gate with the same expenditure. In connection with Fig. 65 this hysteresis is characterized for a carrying charge by the difference of pressures, expressed by cut b.

By the reason, which lead during a change in the expenditure to the disturbance/breakdown of the stability of pressure (and to the appearance of hysteresis), are characteristic of spring and friction of the movable parts of valve, and also a change in those acting during the lifts of the gate of the valve (in transient conditions) of the forces of pressure of liquid, including force of inertia and of flow forces.

For an increase in the stability of pressure during a change in the expenditure first of all, it is necessary to raise the elasticity of spring, which is reached by an increase in its length (by decrease in the coefficient of its hardness C), and also maximally to reduce frictional force and the course of gate.

The dependence of pressure on the expenditure ideal from this viewpoint of valve is expressed not allowing for flow forces of the

vertical straight line a (Fig. 65) both with an increase of the expenditure from the zero to the maximum value and during a reduction in it from the maximum to the zero. Pressures at the end of the lift, and also at the end of the coverage of gate for this valve coincide with the pressure of the aaaa of the beginning of lift.

The frictional forces of plunger determine the sensitivity δ valve to pressure change, which is estimated at relation

$$\delta = \frac{\Delta p}{p},$$

where Δp - the excess of pressure above the nominal (calculated) pressure p, determined by the force of valve spring. Sensitivity oscillates within sufficiently wide limits (δ = 0.03-0.1) and depends on the construction of the working section of gate and form of saddle (plane, conical), and also from friction of plunger. Thus, for instance, in valves with plunger gate (see Fig. 67) sensitivity lower than in valves with conical, but fact more with edge gate.

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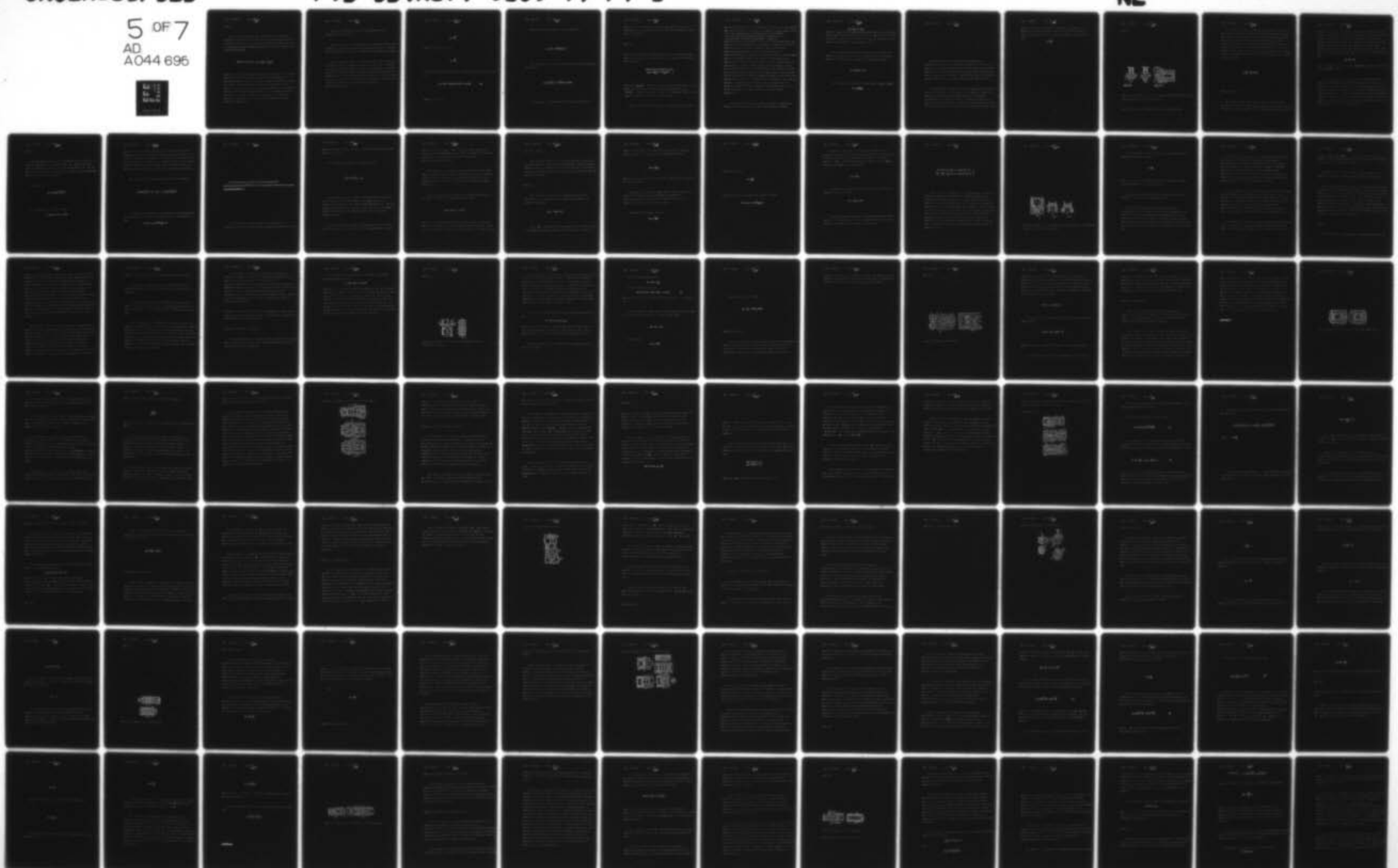

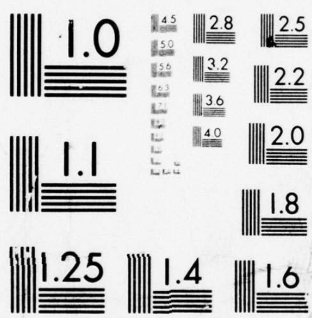

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Taking into account frictional forces the force condition of equilibrium (not allowing for flow forces of fluid flow and force of inertia), which act on the conical gate of valve with sharp edges in the beginning of valve opening (breakaway of gate from saddle) or of coverage $(Q \approx 0;\ h \approx 0)$,

$$\Delta p f_0 = P_0 \pm R = Ch_0 \pm R; \quad \Delta p = \frac{P_0 \pm R}{f_0} = \frac{Ch_0 \pm R}{f_0},$$

where P_0 is a force of spring with zero valve lift; Δp - the pressure differential of liquid, which corresponds to the beginning of discovery/opening or to the end of the coverage of the gate of valve; $f_0 = \pi d^2/4$ - the sectional area of the gate of valve on line of contact of its with the edges of saddle (projection of the surface of valve, washed by liquid under pressure); R is a frictional force of the rest of the moving elements of the valve; C - the stiffness coefficient of spring; h_0 - the preliminary reduction of spring (during zero valve lift).

During the approximate practical calculations friction frequently they disregard.

After the gate is detached away from its seat/socket (it will be discovered), the pressure differential will change due to a decrease in the effective area of gate, on which acts the pressure of liquid.

From the chalchulation of the diagram, given in Fig. 63a and b, it follows that with enclosed valve with the sharp edges of saddle the pressure of liquid acts on its gate over section with a diameter of d, whereas with that which was opened valve this section will be determined by the alternating/variable diameter $d_1 < d$ of the section of the cone of the gate. In accordance with this, the effective area of valve, on which acts the pressure of liquid after discovery/opening gate,

$$f_{эф} = \frac{\pi d_1^2}{4}$$

will be less than the area

$$f_0 = \frac{\pi d_0^2}{4},$$

that corresponding to enclosed gate. Effective area (see Fig. 63a and b)

$$f_{эф} = \frac{\pi}{4}(d - 2h \sin\beta \cos\beta)^2 = \frac{\pi}{4}(d - h \sin 2\beta)^2, \qquad (40)$$

where $\beta = 90° - \alpha/2$.

Flow passage cross-sectionals area of valve slot

$$f_{щ} = \pi \left(d - h \frac{\sin 2\beta}{2}\right) h \cos\beta.$$

After substituting the area of slot into (38) of flow equation, we will obtain

$$Q = \mu\pi \sqrt{\frac{2}{\rho}} \left(d - h \frac{\sin 2\beta}{2}\right) h \cos\beta \sqrt{\Delta p},$$

It is obvious, for valves with the small lift of gate the

decrease in the effective area of the $f_{э\phi}$ of valve, caused by difference $d - d_1$, negligibly little and it in the majority of cases can be disregarded.

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The force condition of equilibrium, which act on the gate of valve with the sharp edge of seat/socket, with the maximum fluid flow rate takes the form

$$\left.\begin{aligned}\Delta p_{max} f_{э\phi} = P_{max} \pm R = C(h_0 + h) \pm R;\\ \Delta p_{max} = \frac{P_{max} \pm R}{f_{э\phi}} = \frac{C(h_0 + h) \pm R}{f_{э\phi}},\end{aligned}\right\}$$

where the $P_{max} > P_0$ - force of spring during its compression on $h_0 + h$; here P_0 is a force of spring during precompression on h_0; $f_{э\phi} < f_0''$ - the effective area of valve [see expression (40)]; h is a valve lift along the axis (see Fig. 63a and b).

Effect of the form of seat/socket. In real valves seat/socket

has not sharp edges, and a certain surface (see Fig. 63c), in view of which the stability of the forces of pressure of liquid, which act on valve and, consequently, also difference in the pressures $p_н$ at the start of the opening and p_0 at the end of the closing will be even more considerable. Figure 63c shows that before the breakaway of the gate of valve from saddle the force of spring is balanced by the pressure of liquid, which acts on the projection of the washed surface of gate, by which for a hermetically sealed valve will be the sectional area of opening/aperture with a diameter of d. After as gate is detached away from its seat/socket, liquid will enter the slot, formed by the saddle and the cone of gate, as a result the area to which will act the pressure of liquid, it will increase by the projection of the area of saddle by the plane, perpendicular to the axle/axis of gate. It is obvious that the pressure of the inward flange of the contact of gate with saddle is equal to operating pressure p_1, whereas of the outward flange of slot it is lowered up to a pressure of p_2, equal to pressure at output/yield from valve. With the conicity of the surfaces, forming the slot, a change in the pressure from p_1 to p_2 occurs according to the law, depicted on Fig. 63c (shaded area/sites).

In accordance with this, the force condition of equilibrium, acting on the gate of valve in moment the coverage of window,

$$P_{np} = \Delta p f_0 \pm R + p_{cp} f_{2H}$$

where the p_{cp} are the mean pressure, which acts on this band after the breakaway of valve from saddle (h > 0); $f_{2H} = \pi (D^2 - d^2)/4$ - the projected area of the surface of the band of seat/socket to the plane, perpendicular to the axle/axis of gate.

For determining the supplementary force of the pressure of liquid, which acts in the slot in question, they use the average value of the pressure which according to experimental data

$$p_{cp} = 0{,}45\,(p_1 - p_2).$$

Hence the pressure at which the valve will be closed $(h \approx 0)$,

$$p_0 = \frac{P_{np}}{f_0 + 0{,}45 f_{2H}}.$$

With the insufficient airtightness of conical gate, supplementary force from the pressure of liquid in the slot of the seat/socket of valve will enter also into the balance of the forces, which act at the torque/moment of its breakaway from saddle during valve opening, in view of which a similar valve will be discovered at pressure lower than the pressure, obtained from expression (39).

Discontinuity in pressures it began discovery/openings also of the end of the closing of this valve it is possible to lower with a decrease in the width of the bearing surface of seat/socket. Specifically, the contact of the gate of valve with saddle on the edges, close to acute/sharp, is provided by the frequently fact that the apex angles of gate and seat/socket are fulfilled different (Fig.

66a). The area f, on which acts the pressure of liquid in the beginning of discovery/opening and at the end of the closing of valve, is determined for this valve by the area of the apex of the cone

$$f=\frac{\pi d_2^2}{4}.$$

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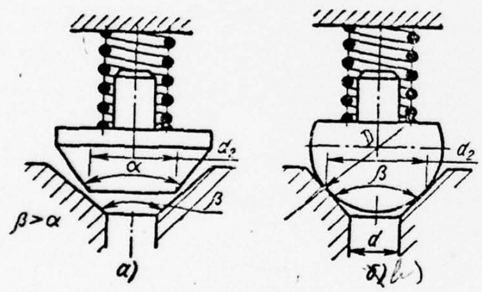

Fig 66.

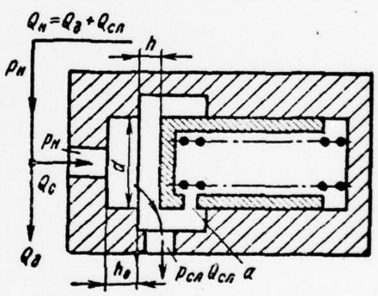

Fig. 67.

Fig. 66. The valves: a are c) by the unequal angles of tapers of gate and seat/socket; b) with spherical gate.

Fig. 67. The design diagram of plunger type overflow valve.

Are common also mushroom valves and spherical gate (Fig. 66b). These valves possess the relatively small flow resistance of liquid (1.5-2 times it is lower than in valves with conical gate). Angle β the saddle of the last/latter valve is usually equal to 90° and diameter D of sphere D = 2d. The area on which acts the pressure of liquid in the beginning of discovery/opening and at the end of the closing of this valve, is the sectional area of sphere on the points of its contact with seat/socket by the plane, perpendicular to valve spindle. This area

$$f = \frac{\pi d_2^2}{4} = \frac{\pi}{4} D^2 \sin^2 \beta_2.$$

Overflow valves.

The examined valves can be applied both as protective with occasional action and as overflow, that support the constant pressure of liquid on system by means of the continuous diversion/tap (gutter)

of part of the liquid in tank. In practice this hydraulic apparatus is called frequently the valve of a jump/drop in the pressure or by the valve of pressure, intended for maintaining the determined pressure differential in that which supply and that which discharge hydraulic lines. The hydraulic parameters of this valve are a pressure difference in of pressure $p_н$ and the drainage $p_{сл}$ lines and the expenditure (bypass) of $Q_{сл}$ into the line of gutter (Fig. 67):

$$Q_{сл} = Q_н - Q_д,$$

where the $Q_н$ - the feed of pump; $Q_д = Q_н - Q_{сл}$ - the expenditure of user (hydraulic engine).

On the strength of the specificity of the work of overflow valves, they usually are fulfilled with plunger gate. Value h_0 the overlap by the gate (plunger) of the window through which the oil after valve opening overflows into tank, must be somewhat more than the spread/scope of the possible longitudinal vibrations of plunger during its vibrations in order that it would not be hit in this case against its support. For the attenuation of vibrational energy in valve provided for choke opening/aperture a.

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Communication/connection between pressures at the inlet into the valve of $p_{\text{н}}$ and at the output/yield of p_{CA}, and also by fluid flow rate through the valve (by expenditure for gutter) of Q_{CA} we will obtain by the joint solution of the following equations (by flow forces and friction we disregard):

fluid flow rate

$$Q_{CA} = \mu\pi\, dh \sqrt{\frac{2\,(p_{\text{н}} - p_{CA})}{\rho}};$$

the equilibrium of the gate of valve

$$P_0 + C(h_0 + h) = (p_{\text{н}} - p_{CA})\frac{\pi d^2}{4},$$

where d and h we disregard diameter and the course of gate (valve opening); P_0 - the force of the precompression of spring (with $h + h_0 = 0$; C is a stiffness coefficient of spring; h_0 is a size/dimension of the overlap by the plunger of the window of gutter in the enclosed position of valve, i.e., the size/dimension to which must move the gate from its support to the position of the beginning of bleeding.

After solving these equations relative to h, we will obtain

$$Q = \mu\pi d\left[\frac{\pi d^2}{4C}(p_{н} - p_{сл}) - \frac{P_0}{C} - (h_0 + h)\right]\sqrt{\frac{2(p_{н} - p_{сл})}{\rho}}.$$

The pressure differential in the beginning of discovery/opening the flow area (at the moment of separation of gate from saddle) of valve

$$\Delta p = (p_{н} - p_{сл}) = \frac{4[C(h_0 + h) + P_0]}{\pi d^2}.$$

~~To the pressure differential in the beginning of discovery/opening the flow area (at the moment of separation of gate from saddle) of valve~~

To overflow valves is not presented the requirement for airtightness; therefore the force, which ensures airtightness, in the

enclosed position of valve, it can be accepted equal to zero and with respect P_0 = to 0.

In this case the equilibrium condition of valve

$$C(h_0+h)=\frac{\pi d^2}{4}(p_H-p_{CA}).$$

It is obvious, and in the case of overflow valve for obtaining it is possible the more flat curve of $p_H=f(Q_{CA})$, i.e., for a decrease in the degree of the effect of the flow rate of the Q_{CA} of liquid for the pressures of p_H, one should decrease the stiffness coefficient of spring C and increase diameter d of the passage opening of valve.

Action of flow forces. After the breakaway of gate from saddle (h > 0) will appear in the place of the throttling/choking of liquid

flow forces of the $P_{гидр}$, which attempts to close valve, i.e., acting in the same direction, as the force of spring. This force can be considered as supplementary hydraulic spring with alternating/variable hardness.

Flow forces is the reaction of fluid flow to the gate of valve and can reach the value, capable significantly to change the balance of the acting on it forces. In certain cases the force of spring comprises in this balance less than 50o/o of overall force, which acts on gate.

The axial component of flow forces of fluid flow according to the law of a change in the momentum (see Fig. 63b)

$$P_{гидр} = Q\rho\,(u_1 - u_2 \cos \alpha/2),$$

where Q and ρ - volumetric flow rate and the density of liquid; u_1 and u_2 - the average speed of the liquid before the gate (in feeder) and in the passage slot (in jet) of valve; $\alpha/2$ - the airflow angle in

the slot of valve.

Investigations show that the flow direction for the widespread in valves angles of taper of gate with apex/vertex (<140°) virtually coincides with the generatrix of the cone of the gate. In accordance with this, the angle $\alpha/2$ can be accepted equal to semivertex angle of the cone of the gate.

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Since speed $u_1 \ll u_2$, it in the majority of cases of the calculation can be disregarded. As a result we will obtain the simplified expression

$$P_{\text{гидр}} = -Q\rho u_2 \cos \alpha/2.$$

Since $P_{\text{гидр}}$ increases with an increase in the flow rate, and consequently, it increases with the lift of the gate of valve, in

practice frequently is introduced by analogy with the concept of spring constant C the concept of the hydrodynamic hardness:

$$C_{\textit{гидр}} = \frac{\Delta P_{\textit{гидр}}}{\Delta h},$$

where Δh is the increase in the valve lift, caused by an increase in the fluid flow rate.

Experiments show that the $C_{\textit{гидр}}$ just as spring constant C, changes over a wide range of flow rates proportional to discovery/opening gate and in practice does not depend on the pressure differential on the gate of valve.

Summarizing hydrodynamic hardness

$$C_{\textit{гидр}} = \frac{\Delta P_{\textit{гидр}}}{\Delta h}$$

with spring constant

$$C=\frac{\Delta P_{np}}{\Delta h},$$

we will obtain the resulting total hardness of valve

$$C_{pes}=C_{гидр}+C=\frac{\Delta P_{гидр}+\Delta P_{np}}{\Delta h}.$$

Experiment is show that the hydrodynamic hardness in many instances exceeds (2 times and more) spring constant, in view of this an increase in the force on valve Δp, the caused by the total hardness $C_{рез}$, considerably exceeds the increase in the $\Delta P_{пр}$, caused by the hardness of spring itself:

$$\Delta P \gg \Delta P_{пр}.$$

In accordance with this a change in the force of pressure of liquid on the gate of valve

$$\Delta P = \Delta P_{гидр} + \Delta P_{пр}.$$

Taking into account flow forces of equation in question, which express the equilibrium of the gate of valve with the sharp edges of saddle, they accept with the maximum flow rate form

$$\left.\begin{aligned} P_{\max} &= \Delta p_{\max} f_{э\phi} = C(h_0+h) + Q\rho u_2 \cos\frac{\alpha}{2} \pm R; \\ \Delta p_{\max} &= \frac{P_{\max}}{f_{э\phi}} = \frac{1}{f_{э\phi}}\left[C(h_0+h) + Q\rho u_2 \cos\frac{\alpha}{2} \pm R\right]. \end{aligned}\right\}$$

Effect of force of inertia. To the characteristic of valve in transient condition, affects also its dynamics, caused by the acceleration of moving elements. The inertia forces in valve are determined by acceleration and the mass of gate with the apparent additional mass of the spring whose value usually is accepted equal to 1/3 masses of spring. In certain cases (in large-size valves, and also at small section and the large length of drain channels) is considered also the mass of the liquid above the valve and in channels. For the approximate computations the apparent additional mass of spring and liquid in this case usually accept equal to 0.5 masses of spring.

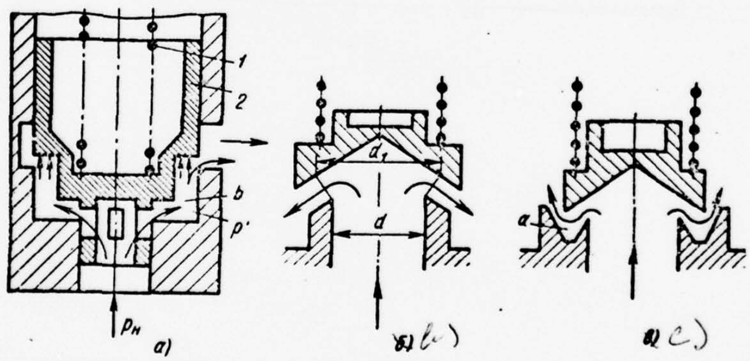

Fig. 68. Diagrams of the cell/elements of valve with the compensation for the forces, which act on gate.

The acceleration of gate is accepted of the condition of its uniformly accelerated motion

$$J=\frac{2h}{\Delta t^2},$$

where h and Δt - the height/altitude and the duration of ascent (discovery/opening) of the gate of valve.

Experiment shows that the pressure overshoot during valve opening can reach 50o/o of nominal pressure.

Methods of stabilization of pressure. For pressure stabilization, it is necessary that after discovery/opening gate would arise the additional force, which would load it in the direction of the effect of pressure of liquid (it would compress spring). For this purpose frequently is utilized the action on the gate of fluid flow during a change in the direction of the entering jet.

Figure 68a shows the diagram of the plunger valve in which for this purpose is executed the intermediate annular chamber b, arrange/located after passage slide-valve slot. In this chamber in the work of valve, is formed intermediate pressure $0 < p' < p_{н}$, which creates supplementary force on plunger gate 2, which counteracts to the force of spring 1. By means of the appropriate selection of the area of this chamber it is possible to attain the required correction of the characteristic of valve.

For an improvement in the characteristic in question are applied also the valves with reverse/inverse cone (Fig. 68b), in which because of the considerable deflection of fluid flow it is possible to obtain the flow forces, capable to partially compensate for the growing with valve lift force of spring. Besides this the compensation effect is here caused also by the fact that with the lift of gate increases its effective area, since during lift $d_1 > d$.

The compensation effect can be raised during the application/use of the dual conicity of landing seat/socket (Fig. 68c) because of which in the intermediate chamber a with the open gate is created

certain pressure $0 < p' < p_{н}$, which affects the gate in the direction of the effect of pressure of liquid (against the direction of the action of the force of spring).

Experiment shows that in the valve of this diagram it is represented possible to obtain virtually stable characteristic.

Oscillations (vibration) of valves. On the gate of valve, which is located in fluid flow, constantly acts the fluctuating pressure of pump, which is the periodic function of time with the periods, equal to the revolution of the rotor of pump. Since the valve is the dynamic system, connected with elastic medium - liquid, in this system under specific conditions, can arise the auto-oscillations, which are capable to upset the operation of the entire connected with hydraulic valve (to cause the pulsations of pressure, etc.), and also to cause a breakage in the valve spring.

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Under the known conditions valve, in particular valve with the

conical edges of saddle (see Fig. 63c), it can enter into transient conditions into oscillations (vibrations), which under conditions of resonance will cause considerable fluctuations of pressure in an entire hydraulic system. Thus, for instance, with an instantaneous increase in the flow rate the gate of valve on the strength of the action of force of inertia will arrive to motion (it will be discovered) with certain delay, as a result upstream pressure sharply it grow/rises, which will derive gate beyond the limits of the required position of equilibrium, which corresponds to new flow rate. This excessively large discovery/opening (displacement/movement) gate cause a sharp decompression before it, which, in turn, will lead to the excessively large displacement/movement of gate to the side of coverage.

Furthermore, in valve with conical saddle the fluctuations of flow rate and their accompanying speed fluctuations of fluid flow in the slot between the gate and the saddle produce fluctuations of the pressures in it, which as a result of the disequilibrium of the forces, which act on gate, are the supplementary factor, which excites fluctuations. It is obvious, the higher the pressure differential in valve and the greater the width of the edge of saddle, i.e., the greater the difference D - d, where D and d are the diameter of basis/base and apex/vertex of conical saddle (see Fig.

63c), the large will be the examined/considered exciting effect.

As a result of indicated, the gate of valve can enter the auto-oscillations, which proceed usually with high amplitude and frequency.

The source, which excites the fluctuations of valves, can be also the other external and internal disturbance/perturbations, by basic from which it is the flow fluctuation of liquid, applied pump [7].

For a reduction in the probability of the entry of valve into resonance oscillations, one should avoid the mode/conditions in which the frequency of the perturbation momentum/impulse/pulses coincides with the natural vibration frequency of the gate of valve, determined by the mass of of gate itself and spring with the connected volume of liquid, or is multiple by it. It is necessary also to avoid the agreement of the natural vibration frequency of valve with the ripple frequency of fluid flow in system.

The resonance phenomena can be removed by the creation of resistance in moving the gate of the valve whose force would be as much as possible proportional to the speed of its displacement/movement. These requirements most completely satisfies hydraulic damping (see Figs. 63a and 67), with the aid of which it is possible to ensure stability during all virtually possible excitations. It is obvious that the damping effect of valves depends on the size/dimension of the throttling channel which usually is selected experimentally[1].

FOOTNOTE [1]. The analysis of the transient stability of valve is the theme of course the "theory of automatic control and the dynamics of hydropneumatic systems." ENDFOOTNOTE.

Lamellar (flat/plane) type valves.

In some constructions of hydraulic drive, are applied the valves with flat/plane seating bosses (Fig. 69a), which differ in terms of high airtightness and reliability.

To the gate of valve from the side of liquid, acts during discovery/opening the force

$$P = \Delta pf + Q\rho\,(u_1 - u_2 \cos\beta),$$

where $\Delta p = p_1 - p_2$ - the pressure differential to (p_1) and afterward (p_2) gate; $f = \pi d^2/4$ - the area of the seat/socket (feeder) of valve; $Q\rho$ (u_1 is $u_2 \cos\beta$) - the flow forces (reaction of flow to gate), caused by a change in the momentum; here Q is volumetric fluid flow rate through the valve; ρ - the density of liquid; u_1 and u_2 - the average speed of the liquid before the gate (in the opening/aperture of seat/socket) and in the slot of valve; β is an angle of deflection of the jet, which escape/ensues of the slot of valve.

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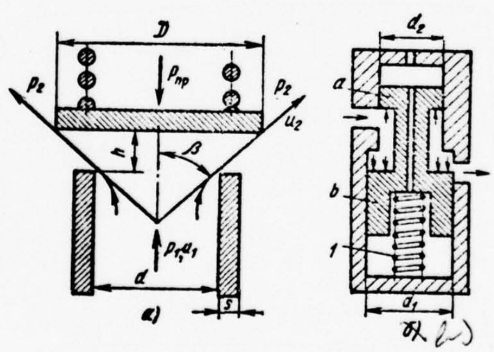

Fig. 69. The valves: a are c) by flat/plane gate; b) with differential plunger.

The airflow angle β is the value of variable and depends on lift h of the gate of valve, decreasing with an increase in the latter, and also from size/dimensions of D, d and s (Fig. 69a), in view of which the precision determination of the dependence of this angle on lift to determine is extremely complicated; therefore they are restricted to approximate estimates and the experimental data. Under condition D = d of this angle can be taken as during the approximate computations β = 69°. During considerable overlaps (D > 1) and small lifts of gate it is possible to accept β = 90°.

The force of spring, which acts on gate after valve opening (h > 0),

$$P_{np} = P_0 + Ch = (h_0 + h) C,$$

where P_0 is Ch_0 the force of initial compression of spring (with h = 0); here h_0 - initial compression of spring (with h = 0); h is the variable value of the lift of gate, which ensures flow rate of Q.

In accordance with this the condition of the equilibrium of the gate of the valve:

before discovery/opening gate

$$P_0 = Ch_0 = \Delta p_0 f;$$

after discovery/opening gate

$$P_{np} = (h_0 + h)\, C = \Delta pf + Q\rho\, (u_1 - u_2 \cos\beta), \qquad (41)$$

where Δp_0 and Δp are the pressure differential with h = 0 and with h > 0.

With low flow rates (with small h) by the reaction force of flow it is possible to disregard, as a result we will obtain

$$(h_0 + h)\, C = \Delta pf.$$

Consequently,

$$h_0 + h = \frac{\Delta pf}{C}.$$

From equation (41) it follows

$$\Delta p = \frac{P_{np}}{f} - \frac{Q\rho\,(u_1 - u_2 \cos\beta)}{f}.$$

Differential valves.

For the size decrease of springs and forces of their tightenings which at large flow rates and the pressures of liquid take in the case action of direct valves inadmissible values, are applied differential valves with the hydraulic balancing of part of the

force, developed with the pressure of liquid. This balancing in the majority of constructions is realized with the aid of supplementary piston 1, connected with the basic breech screw of valve.

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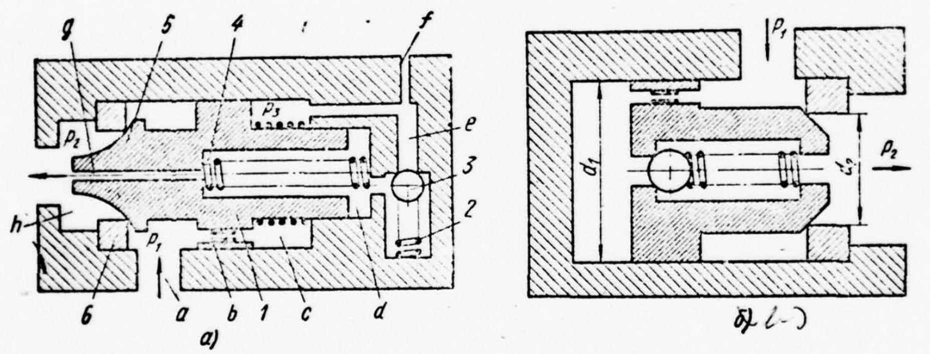


Fig. 70. Two-stage safety valves.

Virtually as the basis of the constructions of the majority of differential type valves (Fig. 69b) is placed the unbalanced plunger, shouldered a and b of different diameters. It is obvious, in this valve spring it receives only the force of the pressure of liquid, which acts on the effective area, equal to a difference in the areas of the end/faces of the plunger:

$$\Delta f = f_1 - f_2 = \pi/4\,(d_1^2 - d_2^2).$$

The force of the precompression of spring 1 for this valve find from equation

$$P_0 = p\,(f_1 - f_2) = p\,\frac{\pi}{4}\,(d_1^2 - d_2^2),$$

where d_1 and D_2, the diameters of bands b and a of inner valve,.

The excessive decrease in the effective area of the gate of

valve, i.e., a decrease in the difference in the areas (f_1 - f_2) of bands a and b, will lead to the fact that the fraction of frictional forces in the balance of the forces, which act on plunjer, will be so great that the valve will not be able satisfactorily to fulfill its function due to large hysteresis of friction (see Fig. 65).

Two-stage safety valves.

During the application/use action of direct valves on high-pressure, systems the diameters of their gates are virtually limited by size/dimension 25 mm, since at their higher values inadmissibly increase the forces of springs.

For a decrease in the force of spring at given flow rate and pressure, and also for an increase in the stability of pressure are applied the two-stage valves (valves with servo effect), shown in Fig. 70a. Liquid under operating pressure p_1 will be fed into chamber a, connected through the choke opening/aperture b with cavity c and cavities c acts on piston 1, holding (together with spring 4) gate 5 in enclosed position. Valve is enclosed until pressure p_3 in cavity c overcomes the force of spring and will not discover pilot valve 3.

After discovery/opening this valve, the pressure of liquid in cavity c as a result of the resistance of the choke opening/aperture b is lowered in comparison with pressure in cavity a, as a result gate 5 is detached away from its saddle and pressure p_1 in cavity a it is lowered to the value at which the fluid flow rate through valve 3 will be equal to that amount of liquid, which will enter cavity c through the choke opening/aperture b. The process of the displacement of liquid and, consequently, also discovery/openings of the basic gate of valve 5 it depends on overflowing into chamber c of liquid from the force main through the choke opening/aperture b.

[redacted]

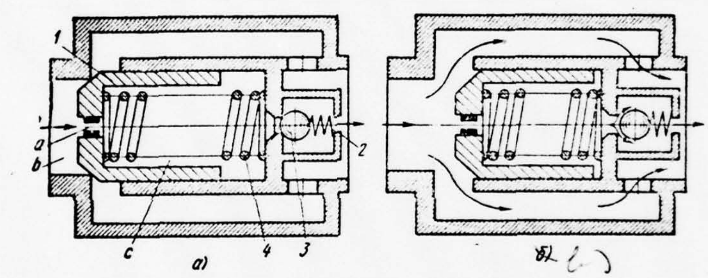


Fig. 71. Schematic of the action of two-stage safety valve.

By a change in the effort/force of the precompression of the spring of 2 gates of pilot valve 3 it is possible to regulate the basic (locking) valve.

For the balancing of gate 5 from the forces of drainage pressure in it is drilled a hole g, connecting the power cavity h of the valve with the cylindrical chamber d whose diameter is equal to the diameter of the saddle of 6 valve.

In the construction of valve, usually is provided for the possibility of remote control of the discharging of pump (by its translation/conversion into the mode/conditions of idling). For this, in valve is carried out opening/aperture f, during connection/compound of which with drain line the pressure in cavity c is lowered up to a pressure of this main line ($p_1 \approx p_2$), as a result gate 5, after moving to the right, will connect pressure and drain lines.

Figure 70b depicts the schematic of a similar valve with auxiliary ball valve within the basic valve. This valve is simple in production; however, it differs in terms of the unbalanced force of

drainage pressure whose value is determined by relation

$$\frac{d_1^2}{d_1^2-d_2^2},$$

where d_1 and D_2 are a diameter of the breech screw and seat/socket of valve.

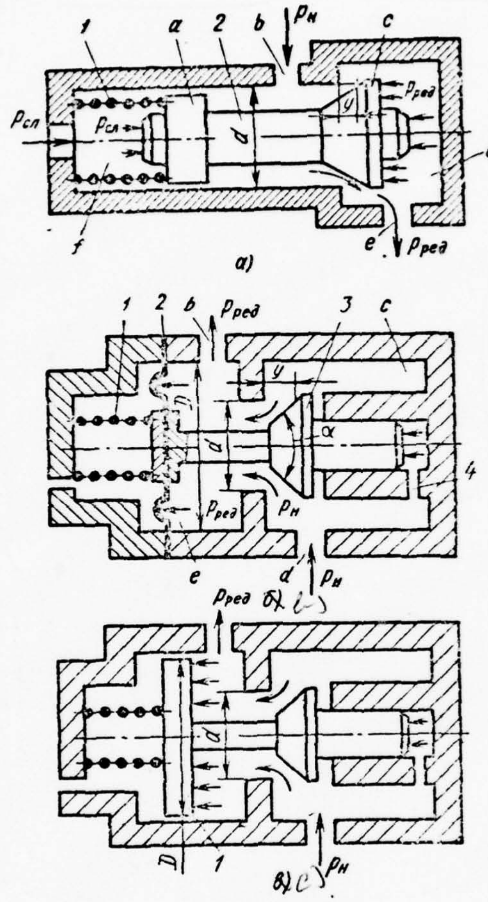
The examined valve frequently is carried out through the schematic, presented in Fig. 71. At pressure in system than lower assigned (Fig. 71a) the gate of ball bearing valve- sensor 3 is enclosed. In this case, pressure in cavities b and c which communicate through the choke opening/aperture a in piston 1, are equal. Spring 4 holds piston 1 in the position by which inlet duct b is enclosed.

During a pressure increase higher than rated value for which is designed spring 2, the ball bearing gate of 3 valve- sensors is open/disclosed, and pressure in cavity c falls, as a result of which in cavities b and c it is created a jump/drop in the pressure, under

action of which the piston of 1 overflow valve is moved, connecting the channel of forcing with tank (Fig. 71b).

For the smoothing (cutting) of the pressure overshoots (for example the pressures, which develop with hydraulic impact), one should apply action of direct valves (see Fig. 62), since during application/use for these purposes of valves with servo effect (see Fig. 70-71) can arise as a result of unavoidable delay in the final adjustment of signal (in discovery/opening the basic gate of valve) the large pressure overshoots. As can be seen from the schematic, given in Fig. 71, the displacement of the basic gate (overflow valve) can occur only after will be opened pilot valve and liquid, filling chamber c, will be extrude/excluded into the tank through the opening/aperture valve sensor 3. However, these valves differ higher than their single-stage types, by the stability of pressure which is reached here because of the fact that load on gate 5 (see Fig. 70) valve it is realize/accomplished by the pressure of liquid, maximum value of which is determined by the characteristic of spring 2 of pilot valve 3 of small size/dimension.

Fig. 72. Calculated schematics of reduction valves.



Because of the small fluid flow rate through the choke opening/aperture b in piston, the pressure of liquid on piston 1 during a change in the flow rate practically will not change, and consequently, stable will be pressure p_1 during all flow regimes of the liquid through the working window (slot) of overflow valve.

Reduction valves of constant pressure.

Reduction valve or reducer (Fig. 72a) is the automatically acting throttle/choke whose resistance is equal at each this torque/moment of the difference between the variable pressure of $p_н$ at the entrance into valve and the constant (reduced) pressure of $p_{ред} < p_н$ at output/yield. Valve is intended for lowering (reduction) in the pressure in any discharge section of main line (hydraulic lines) and of maintaining this pressure by constant independent of the pressure in supply main, which are must only several (to 2-3 kgf/cm²) to exceed the reduced pressure.

These valves are applied in essence in such a case, when from one source of flow rate (pump) are supplied several users (servomotors), that require different pressures. The source of flow

rate (pump) in this case they rely on the maximum pressure, necessary for the supply of any of the users.

In the simplist form reduction valve (Fig. 72a) it is plunger 2 with the throttling conical knob/cap c at the right end and with the balancing small piston a on left. Liquid under the high pressure of $p_н$ will be fed to channel b and is abstract/removed under the reduced pressure of the $p_{ред} < p_н$ through channel e. Decompression from input $p_н$ to exit $p_{ред}$ and maintaining the latter at fixed level is caused by dynamic equilibrium of forces, which act on mobile/motile plunger 2, from which the effort/force of spring 1 acts to the side of an increase in discovery/opening passage slot by height/altitude y, which connects channels b and e, but the pressure of $p_{ред}$ in chamber d and flow forces act to the side of a decrease in this slot.

At certain low (it is less than calculated) pressure of the $p_{ред}$ of plunger 2 with force of spring 1 is wrung out and increases to the right clearance y, through which liquid enters from channel high-pressure b of $p_н$ into channel e of the reduced pressure of $p_{ред}$.

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After the pressure of $p_{ред}$ in the last/latter line will exceed the design pressure to which are adjusted spring 1, plunger 2 under effect of pressure the $p_{ред}$ of liquid will move to the left, partially or completely overlapping the admission of liquid from channel b into channel e of the reduced pressure.

When the diameter of the section of the conical knob/cap c of gate by the plane, which passes on contact points of its with the edges of saddle (it corresponds to saddle with sharp edges), is equal to the diameter of small piston a, of the force of pressure of $p_н$ on plunger 2 in the beginning of discovery/opening slot (clearance y = 0) they are balanced ($p_{ред}'$ does not depend on $p_н$), and the equation of state of valve takes the form (forces of the inertia of friction we disregard)

$$p_{ред}f = P_0 = y_0C;\ p_{ред} = \frac{y_0C}{f},$$

where f = $\pi d^2/4$ - the area of the indicated section of the cone of the gate; $P_0 = Cy_0$ - the effort/force of the precompression of spring 1 (with y = 0); y_0 and C are the precompression of spring and its hardness.

With the open slot (y > 0) to gate will additionally act to the side of the coverage of gate the flow forces of the P_e, taking into account which the equation of the equilibrium of plunger 2 will take form

$$p'_{ped} f = C(y_0 + y) - P_e;$$
$$p'_{ped} = \frac{C(y_0 + y)}{f} - \frac{P_e}{f},$$

where the p'_{ped} is the reduced pressure with y > 0.

During small displacement/movements values y P_z can be as a result of their relative smallness disregarded, as a result for the calculation of the reduced pressure it is possible to use the preceding/previous equation which shows that with the adopted assumptions the computed value of $p_{ред}$ does not depend on the inlet pressure of $p_н$. However, as a result of the instability of effect on flow forces of the P_z of the pressure differential of $\Delta p = p_н - p_{ред}$, is observed also certain disturbance/breakdown of the stability of $p_{ред}$, i.e., $p_{ред} = f(p_н)$.

For the compensation for effect on the $p_{ред}$ of the possible changes of the pressure of $p_{сл}$ in the drain line of hyiraulic system, the latter is connected with chamber f, in view of which the force of drainage pressure on small piston a of plunger 2 is summarized with force of spring 1.

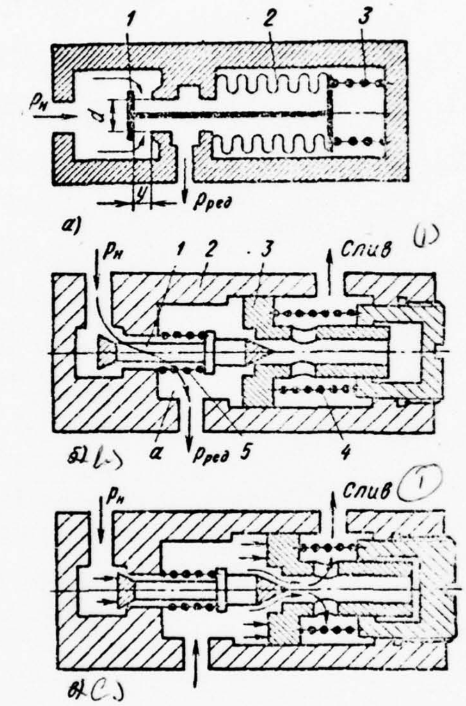
A deficiency/lack in this valve is low sensitivity to changes in the $p_{ред}$, caused by friction of piston and by small area of the cell/element on which acts the reduced pressure. For the elimination

of friction and increase in the sensitivity in low (3-5 kgf/cm^2) reduced pressures, are applied the valves, the role of piston in which it performs rubber-fabric corrugated diaphragm 2 (Fig. 72b).

Liquid under the high pressure of the $p_н$, applied through channel d into chamber c, in passing by the throttling slot by height/altitude y, formed by conical gate 3 and by the seat/socket of valve, enters chamber e and channel b of the user of the reduced pressure of $p_{ред}$. Spring 1 as in the schematic examined above, attempts to discover the gate of 3 valves, but the forces of pressure of the $p_{ред}$ of liquid on diaphragm/membrane 2 and the forces of pressure of $p_н$ on connected with it gate 3 attempt it to close (to decrease the height/altitude of slot y). For the damping of oscillations, is applied throttle/choke 4.

Fig. 73. Membrane/diaphragm (bellows) type reduction valve (a) and reduction-protective valves (b and c).

Key: (1). Gutter.



The expression, which reflects the work of this valve, is based on the following initial equations:

fluid flow rate through the slot of valve

$$Q = \mu\pi\, dy \sin\frac{\alpha}{2} \sqrt{\frac{2\,(p_{\kappa} - p_{pe\partial})}{\rho}}; \tag{42}$$

the equilibrium of gate under the acting on it forces (we allow/assume the even distribution of pressure according to the area of gate and disregard the frictional forces and hydrodynamic effect)

$$P_0 - Cy - \frac{\pi d^2}{4}(p_{\kappa} - p_{pe\partial}) - \frac{\pi D^2}{4} p_{pe\partial} = 0, \tag{43}$$

where y and α - discovery/opening (course of gate) valve and the angle of taper of the gate with its apex/vertex; P_0 is compression of spring with the enclosed gate of valve (with y = 0); D and d are diameters of diaphragm/membrane and valve seat.

After substituting into expression (42) y from expression (43), we will obtain

$$Q = \mu\pi d \sin\frac{\alpha}{2}\frac{1}{k}\left[P_0 - (p_{\kappa} - p_{ped})\left(\frac{D^2}{d^2} - 1\right)\right]\sqrt{\frac{2(p_{\kappa} - p_{ped})}{\rho}},$$

where $k = \frac{4C}{\pi d^2}$.

After relying in equation (43) y = 0, we will obtain expression for determining the maximum pressure of p_{ped} at output/yield from the reducer:

$$p_{ped\,\max} = \frac{P_0}{\frac{D^2}{d^2} - 1} - p_{\kappa}.$$

From the last/latter expression it follows that the outlet pressure of p_{ped} somewhat depends on input p_{κ}, increasing with a decrease the latter.

Because of considerable excess of diameter D of the diaphragm/membrane above diameter d of the seat/socket of valve, and also reduced friction, the examined valve differs in terms of high sensitivity.

At the higher reduced pressures the diaphragm/membrane is replaced by the piston of 1 the same diameter D (Fig. 72c). The calculation is conducted according to those equations with the

addition to expression (43) of the frictional force of piston.

In certain cases it is required to ensure the high sensitivity and maintaining the assigned reduced pressure with low flow rates (close to zero). Since in the plunger circuits examined above with slit gasket and conical gates (Fig. 72) to ensure the required airtightness is difficult, are applied valves with lamellar (plane) gate 1 in which the dynamic seal is realized with the aid of metallic bellows 2 (Fig. 73a).

The equilibrium condition of this valve not allowing for flow forces can be approximately written

$$(p_{\kappa} - p_{ped}) f + p_{ped} F = P_{np} - C_1 y,$$

where $f = \pi d^2/4$ - the cross-sectional area of inlet duct (opening/aperture) with a diameter of d; F is the effective area of bellows (see p. 61); P_{np} - the total effort/force of spring 3 and of bellows 2 during zero valve lift (y = 0); C_1 is the total spring constant 3 and of metallic bellows 2; y is valve opening.

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During a small valve lift by value C_1y it is possible to disregard, as a result we will obtain expression for the calculation of the reduced pressure

$$p_{ред}=\frac{P_{np}}{F-f}-p_{н}\frac{f}{F-f}.$$

reduction-protective valves.

Figure 73b and c depicts the schematic of the valve in which are combined the functions of reduction and safety valves. The position of valve, presented in Fig. 73b, corresponds to supply to the user of liquid under the reduced pressure. In this case, the liquid from the main line of forcing under the pressure of $p_{н}$ enters through the slot between the valve head 1 and the saddle in housing 2 the user.

To that until the pressure of $p_{ред}$ in the system of user achieve/reaches rated value, piston 3 was pulled by spring 4 at extreme left position. In this position the conical neeile of valve 1 abut against the saddle of piston 3, spring 5 is compressed; therefore valve open/discloses the maximum passage of liquid to user.

During a pressure increase of $p_{н}$ at the entrance into reducer, is raised also the pressure of $p_{ред}$ in the cavity of user, as a result piston 3 under the effect of pressure of liquid compresses spring 4 it is moved to the right. In this case, under spring effect 5, is moved to the right also valve 1, as a result the clearance between the left valve head and the saddle of housing decreases. Upon reaching of the assigned reduced pressure of $p_{ред}$ in system, valve 1 will be closed completely. During a decrease in the reduced pressure in system, piston 3 again will move to the left and will discover valve, as a result the pressure in system will increase.

During in the reduced pressure increase over rated value, the force of pressure of liquid on piston 3 increases so, that it, moving

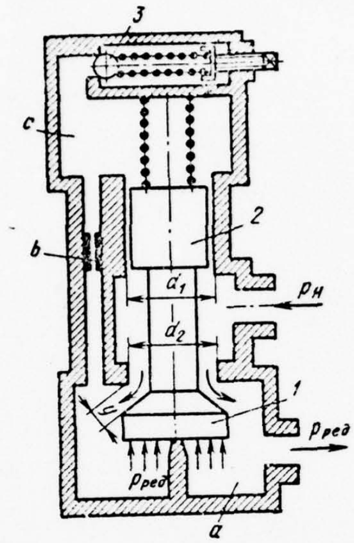
to the right (Fig. 73c), will move away from the conical needle of valve 1, as a result the conical cap of this valve will sit down into its seat/socket of housing 2, and between the needle of valve and the saddle of piston 3 during its further displacement/movement will be formed the clearance through which the liquid from chamber a of the reduced pressure approaches gutter. In this case the reducer acts as the safety valve of the system of user (system of the reduced pressure).

Two-stage reduction valves.

For an increase in the stability of the reduced pressure, are applied the reduction valves of indirect (two-stage) action (Fig. 74). This valve consists of movable conical gate 1, second end of which is carried out in the form of small piston 2. Under the condition of the equality of the diameters of small piston d_1 and of seat/socket D_2 of conical gate and $y \approx 0$, inlet pressure of $p_{н}$, which acts on gate, is balanced. Furthermore, since chamber a of the exit (reduced) pressure of p_{red} is connected through the choke opening/aperture b with chamber c, with $d_1 = D_2$, also is balanced the force of the outlet pressure of the p_{red}, which acts on gate 1.

During increase, the outlet pressure of $p_{ред}$ over computed value ball valve 3 will be opened slightly, as a result the pressure in chamber c is lowered and it will be created the pressure differential between chambers a and c, under action of which the gate of 1 is displaced upward thus decreasing gap y and consequently, lowering the fluid flow rate into chamber a.

Fig. 74. Two-stage reduction valve.



As a result the pressure of $p_{ред}$ will be lowered to the rated value, upon reaching of which ball valve 3 again will be closed, and conical-piston gate 1 will be found in the state of dynamic equilibrium under the effect of pressure of the $p_{ред}$ of liquid.

If the outlet pressure of $p_{ред}$ in chamber a is lowered below computed value, then clearance y under spring effect, which acts on gate, will increase, and pressure in the chamber will be restore/reduced, after being raised to the previous value.

Thus, the flow rate through ball valve 3, determined by the resistance of the choke opening/aperture b, does not depend on the flow rate through clearance y, formed by valve seat and by conical gate 1.

The valve ensures the high stability of the value of $p_{ред}$ practically independent of the inlet pressure of $p_н$ and fluid flow rate from chamber a.

CHOKE ADJUSTERS.

Throttle/choke is the controlling hydraulic apparatus, intended for a change in expenditure and pressure of the flow of working fluid as a result of the passage of this flow through the local resistance. Otherwise throttle/choke is the local adjustable or uncontrolled resistance, establish/installed in the way of flow of liquid for the target/purpose of the creation of a jump/drop in pressure or limitation of its expenditure, attained by the diversion/tap (jettisoning) of part of the liquid through the overflow valve into drain line.

By operating principle, they distinguish:

the throttle/choke of the viscous drag, loss of pressure in which it is determined predominantly by the flow resistance of liquid in the choke channel of large length:

the throttle/choke of the vortex drag, the loss of pressure in which is determined in essence by the deformation of fluid flow and

by vortex formation in the channel of small length.

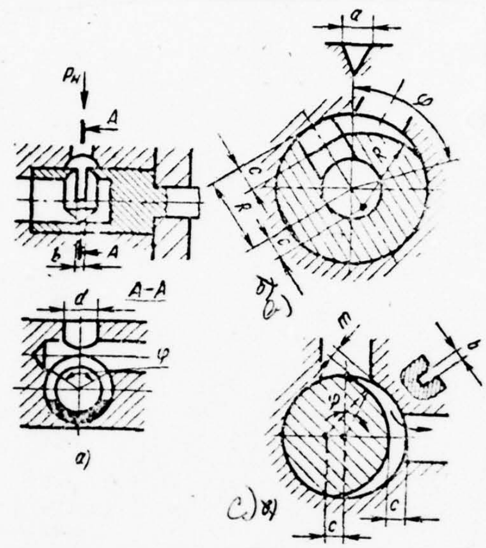
First type throttle/chokes are characterized by large length and small section of choke channel and with respect small Re, in view of which the loss in them of pressure is caused by friction during laminar flow, i.e., the loss of pressure is under otherwise equal conditions the practically linear velocity function of flow of liquid.

A similar throttle possesses the high stability of characteristics whose hearth is understood the stay-put feature of constant/invariable during the repeated settings up of throttle/choke at one and the same regulating position. However, since loss of pressure in this throttle/choke vary directly the viscosity of liquid, hydraulic characteristic its $\Delta p = f(Q)$ depends on temperature. Such throttle/chokes were called the name linear.

In second type, throttle/chokes pressure change occurs practically proportional to the square of the speed of fluid flow, in view of which this throttle/choke is called quadratic. The characteristic of this throttle/choke in practice does not depend on

viscosity in its widespread range.

Fig. 75. Types of crane throttle/chokes.



Plug throttle/chokes.

In low-pressure hydraulic systems (to 50 kgf/cm²) are common throttle/chokes of the type of slewing crane (Fig. 75), the resistance control of which is realize/accomplished by rotation of the plug, performed with different flow areas. By the combination of various forms of flow areas is represented possible to obtain wide range of adjustment and the required characteristic. Specifically, the rectangular form of flow area makes it possible to ensure the in practice linear dependence of expenditure on the angle of rotation of plug.

Figure 75 gives the design diagrams of rotary type widespread plug throttle/chokes. In the throttle/choke whose diagram is given in Fig. 75a, working window is formed by the intersection of the annulus of rotary plug with round opening/aperture in case.

The current area f of the section of window we find from geometric relationship/ratios through formula

$$f = \frac{\pi d}{360} \varphi b.$$

Angle of rotation ϕ plug 0-180°. The maximum sectional area of window corresponds to angle ϕ = 180° and is computed according to expression

$$f_{max} = \frac{\pi d b}{2}.$$

In the throttle/choke whose diagram is shown in Fig. 75b, working window is formed by the intersection of the plug, which has

the eccentric groove/slot (moustache) of triangular form, with round opening/aperture in case. Current sectional area of groove/slot

$$f = ac\frac{\varphi}{90}\sin^2\frac{\varphi}{2}.$$

Angle of rotation ϕ plug 0-90°. The maximum sectional area of groove/slot corresponds to angle ϕ = 90° and is computed according to expression

$$f_{max} = 0{,}5\ ace.$$

In the throttle/choke whose diagram is represented in Fig. 75c, working window is formed by the intersection of the rectangular eccentric groove/slot of rotary plug with round opening/aperture in case. The instantaneous value of the sectional area of groove/slot

$$f = bm = 2cb\sin^2\frac{\varphi}{2}.$$

Angle of rotation ϕ plug 0-90°. The maximum sectional area of groove/slot corresponds to angle of rotation ϕ = 90° and is computed according to expression

$$f_{max} = bc.$$

A deficiency/lack in the throttle/chokes with rotary plug is certain dependence of fluid flow rate through them on the temperature, and also the possibility of the blockage of passage channel, especially at its small sections (with low expenditures and large pressure differentials).

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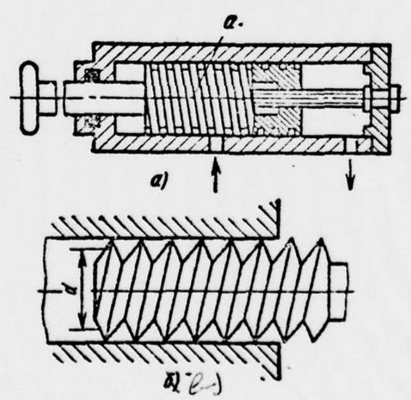


Fig. 76. Diagrams of line choking coils.

Line choking coils.

For the elimination of blockage, are applied the throttle/chokes, the resistance control in which it is achieved by a change in the length of choke channel (line choking coil). In the line choking coil, given in Fig. 76a and b, the resistance is regulated by a change in length a of the groove of single-cut screw/propeller by means of screwing up or unscrewing of screw/propeller. The resistance of this throttle/choke depends on the viscosity of liquid; therefore it can be applied only at the condition of temperature constancy.

Choke channel can be considered as tube rectangular or triangular, depending on the airfoil/profile of thread, section and the calculation of resistance in the first approximation, of news according to common/general/total formulas for ducts. For the groove of rectangular cross section with sides a and b

$$\Delta p = \lambda \frac{L}{4r} \cdot \frac{u^2 \rho}{2},$$

where $L = \pi dk$ - the length of groove with mean diameter d of thread and number k of turns; $r = ab/2(a + b)$ - the hydraulic radius, equal to the ratio of area ab the section of groove to its perimeter 2 (a + b); u is speed of flow; ρ - the density of liquid.

Value

$$Re = \frac{4ru}{\nu}.$$

Quadratic throttle/chokes.

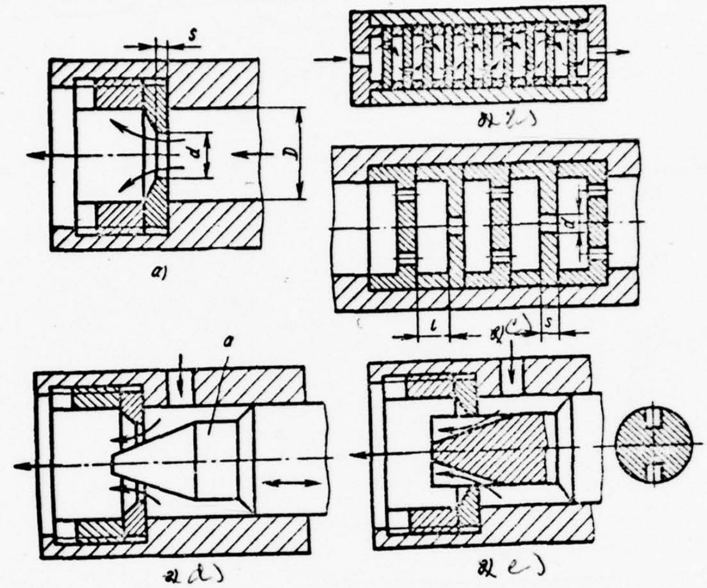
To the throttle/chokes of hydraulic systems, which work under conditions of wide temperature range, is presented the requirement - the form of the flow area of throttle valve must not considerably change the coefficient of fluid flow rate during the measurement of the thermal condition of operation, i.e., during a change in the viscosity of liquid. Furthermore, since during a decrease in the perimeter of the flow area of the channel of throttle/choke decreases the probability of its blockage, this perimeter one should select minimum. It is obvious, better/best from this viewpoint they are throttle/chokes with the less ratio of the perimeter of choke slot to the area of its section and with the shortest channels (passages for a liquid).

These requirements most completely answers quadratic throttle/choke in the form of fine/thin washer (diaphragm) with round opening/aperture and sharp edges (Fig. 77a). The throttling properties of opening/apertures in such washers of caused basically by energy losses during sudden contraction and flow expansion, whereupon flow expansion is accompanied by intense vortex formation in the zone of detached flow. Therefore similar throttle/chokes

possess the minimum dependence of resistance on the viscosity of liquid.

During the development of hydraulic systems, frequently is required the throttle/choke, which possesses high hydraulic resistance (large jump/drop) and stable discharge characteristic during the fluctuations of viscosity. To ensure similar requirements with one choke flange is not represented possible, since the size/dimension of its opening/aperture in this case can be so small, that is possible the blockage by the contaminations of liquid. In view of this are applied multistage throttle/chokes from several consecutive choke flanges (Fig. 77b and c), the operating principle of which is based on repeated contraction and the expansion of the flow of liquid.

Fig. 77. Diagrams of quadratic throttle/chokes.



The resistance of this throttle/choke is regulated with the given size/dimension of opening/aperture by the selection of the amount of washers. Since the distance between washers is usually small, and the cross section of washer (diameter D) greatly in comparison with the section of opening/aperture in washer, it is possible to consider that the hydraulic resistance of this package is caused by losses of pressure during the discharge through opening/apertures in fine/thin wall.

Practice shows that the discharge characteristics of this throttle/choke somewhat they influence the distance between washers (Fig. 77c) which must be not less (3-5) d, where d is a diameter of opening/aperture, or thickness s of the throttling washer or its edge (Fig. 77a) which usually is selected not more (0.4-0.5) d.

Diameter d of opening/apertures in washers must be not less than 0.3 mm, since otherwise is possible their blockage by the contaminations of liquid. With the assembly of choke package, the washers usually are displaced relative to each other so that opening/apertures in them would not be located one against another. With is changed also setting choke package from the alternating consecutively washers with several (two and four) opening/apertures

(Fig. 77c). During the alternation of such washers, the axle/axes of opening/apertures will not be it is located on common/general/total axle/axis.

Are discharged also the single-disk adjustable throttle/chokes. Control here is achieved by the application/use of the choke needle (Fig. 77d), with the aid of which changes the section of choke opening/aperture.

For an increase in the fineness of the adjustment of the adjustable diaphragm throttle/choke its flow area frequently is performed in the form of the angular or rectangular grooves (Fig. 77e), carried out on the barrel of the movable part (lock), which can be carried out both constant and alternating/variable section in the course of this part. This throttle has advantages over the needle-shaped throttle/choke, presented in Fig. 77d both on the possibility of obtaining low expenditures and with the possibility of a decrease in the blockage of slots.

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Loss of pressure in diaphragm throttle/choke with round opening/aperture and sharp edge (see Fig. 77a) is caused by losses by shock [6], in view of which during practical calculations of these throttle/chokes it is possible to apply formula (20) for the calculation of expenditure during the escape of liquid from opening/aperture in fine/thin wall.

The resistance of diaphragm throttle/chokes with the controlling valve/gate (see Fig. 77d) can be calculated from known formula (19) for the calculation of the local losses of head, after accepting coefficient ζ = 2-2.3 (see [7]). These values of coefficient ζ it is possible to accept during the calculation of crane type throttle/chokes (see Fig. 75a), speed of fluid flow for which is assume/taken for the greatest bottleneck of channel.

During the approximate computation of the multi-disk throttle/choke, which consists of n of identical washers, which are located on equal distance l from each other (see Fig. 77b), disregards losses in the chambers between washers and effect on the resistance of counterpressure in them, and also they assume that

total resistance (the pressure differential) the Δp_n of the choke package of washers is equal to the sum of resistance Δp the separate washers:

$$\Delta p_n = \Delta p n \quad \text{или} \quad \Delta p = \frac{\Delta p_n}{n}.$$

Since through each washer flow/lasts per unit time one and the same amount of liquid, is expenditure Q of liquid under the condition of the cross-section equality of opening/apertures in washers

$$Q = \mu\omega \sqrt{2\,\frac{\Delta p}{\rho}} = \mu_n\omega \sqrt{2\,\frac{\Delta p_n}{\rho}}, \qquad (44)$$

where μ - the coefficient of the expenditure for a single washer; $\omega = \pi d^2/4$ - the sectional area of opening/aperture in washer; μ_n are the given coefficient of the expenditure of the choke package of washers.

this coefficient shows, how fluid flow rate through the choke

package, which is of n of identical washers is less with the same the pressure differential, than the expenditure through the throttle/choke with one washer. Its value

$$\mu_n = \frac{\mu}{\sqrt{n}}.$$

From the given data it follows that the diameters of the opening/apertures of throttle/choke with one and many washers with the assigned constants expenditure Q and the pressure differential Dp are connected by relationship/ratio

$$Q = \mu d^2 \sqrt{2 \frac{\Delta p}{\rho}} = \mu_n d_n^2 \sqrt{2 \frac{\Delta p_n}{\rho}}, \qquad (45)$$

where d d_n - the diameter of opening/apertures with one-i by multi-washer throttle/choke.

On the basis of the given data, we can write

$$\mu d^2 = \frac{\mu}{\sqrt{n}} d_n^2; \; d_n = d\sqrt[4]{n}. \qquad (46)$$

During more precise calculations it is necessary to consider the counterpressure of the medium into which occurs the escape of liquid. Under the action of this counterpressure, the actual consumption through the package of washers during turbulent flow can differ from the expenditure, calculated from expression (45).

Experience/experiment shows that of the establish/installed turbulent mode/conditions ($Re > 2 \cdot 10^3$) the actual given coefficient of the expenditure of μ_n' exceeds the calculated μ_n, calculated not allowing for counterpressure, in accordance with which

$$\mu_n^* = k\mu_n = \frac{k\mu}{\sqrt{n}},$$

where k is the correction factor, which characterizes this excess.

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The average value of this coefficient for Re > 2•10³ can be taken as 1.25.

However, in this case it is necessary to consider that, since under given conditions diameter d of sngle-washer throttle/choke less than the diameter of the washers of multi-washer throttle/choke the d_n, different with constant Q and Δp will be also

$$Re = \frac{ud}{\nu}.$$

Value of this number for a multi-washer throttle/choke

$$Re' = \frac{d}{d_n} Re.$$

Taking into account expression (46) this dependence of Re on n will take form

$$\mathrm{Re}' = \frac{\mathrm{Re}}{\sqrt[4]{n}}.$$

The expenditure through the multi-washer throttle/choke depends under otherwise equal conditions on the distance l between washers, optimum value of which one should consider $l \geqslant 5d$.

The calculation of multii-washer throttle/choke with the alternating washers with one and two opening/apertures of equal size/dimensions produces itself on the basis of the condition that the conductivity, according to expression (21), washer with two opening/apertures 2 times is higher than the conductivity of washer with one opening/aperture. In accordance with this the calculation of this throttle/choke it is possible to practically produce on equality (44) with the replacement of the given coefficient of the expenditure of package on

$$\mu_n = \frac{\mu}{\sqrt{n_1 + n_2/2}},$$

where n_1 and n_2 - the amount of washers in accordance with one and two opening/apertures.

In accordance with that which was given, equality (45) will take form

$$d_n = \sqrt[4]{\left(n_1 + \frac{n_2}{2}\right) d}\ .$$

[redacted].

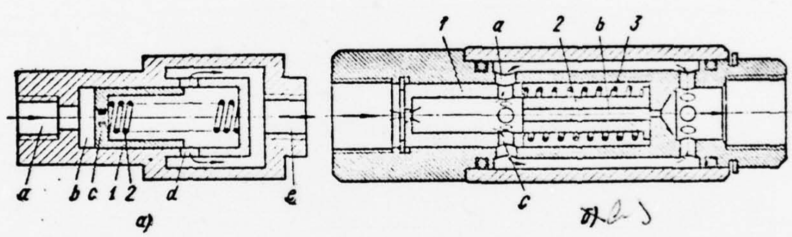

Fig. 78. Diagram and the construction of flow regulator.

AUXILIARY HYDRAULIC APPARATUSES OF CONTROL.

Hydraulic systems include the large amount of different auxiliary hydroaggregates (apparatuses), intended for control, dosing and the flow restriction of liquid, the protection of hydraulic system from the loss of working fluid during the destruction of any aggregate/unit or section of main line, fixing of hydraulic engine in the assigned position and etc.

Limiters (regulators) of fluid flow rate.

When it is required to ensure constant fluid flow rate, for example, for maintaining the constant velocity of the rotation/revolution of hydraulic engine with its alternating/variable load (pressure), in the feed line of user, are istalled the limiters of flow (flow regulators), which by means of the automatic control of loss of head provide the assigned fluid flow rate.

Flow regulator is the controlling hydraulic apparatus, intended for maintaining the determined flow of the passing through it working

fluid without depending on the oscillation/vibration of a pressure difference in that which supply and that which discharge hydraulic lines.

The diagram of a similar flow regulator is depicted on Fig. 78a. Regulator has two throttles one of which the uncontrolled type is executed in base b of piston and second automatic adjusted depending on the pressure differential is executed in the form of window d of alternating/variable section. Liquid from inlet duct a enters chamber b and further through the choke flow-meter opening/aperture c in movable choke piston 1 and windows d of alternating/variable section in housing is directed to toward outlet duct e, connected with user. Piston 1 is loaded by weak spring 2, it will strengthen by which it is balanced the pressure differential, created by the resistance of opening/aperture c. If fluid flow rate by user increases, then will increase a jump/drop in the pressure, as a result piston 1 will move and partially will overlap to the right windows d, decreasing the expenditure/consumption to the value for which is designed the limiter. During a decrease in the expenditure/consumption, piston 1 will move to the left and will decrease the total resistance of windows d and of opening/aperture c.

If opening/aperture c in the base of piston 4 is executed for the diagram, given in Fig. 77a (in the form of opening/aperture in fine/thin wall), then the equilibrium condition of piston 1 will be determined by expression [see also expression (20)]

$$F\Delta p = P_{np};\ \frac{P_{np}}{F} = \Delta p = \frac{Q^2}{f^2}\cdot\frac{\rho}{2\mu^2},$$

where F is a sectional area of piston 1; Δp - the pressure differential in opening/aperture c; P_{np} - the compression of spring 2; Q - fluid flow rate through opening/aperture c; f is an area of opening/aperture c.

Area f and the force of P_{np} design usually for the pressure differential $\Delta p \approx 3\text{-}5$ kgf/cm² for the assigned maximum fluid flow rate.

Figure 78b depicts the construction of the limiter of expenditure/consumption, similar preceding/previous. Unlike the diagram examined above in this construction part 2, the employee of

directing spring 3, has the basic metering hole b, which ensures the minimum fluid flow rate at the maximum pressure by which the expenditure windows a and c are overlapped by floating piston 1 with opening/aperture in end/face.

During a change in the direction of fluid flow, piston 1 is establish/installed in end left position and liquid flow/lasts through completely open windows c and metering hole b. Limiter ensures the virtually stable assigned expenditure/consumption independent of pressure at output/yield (pressure of load) with the error, which does not exceed 1c/o.

The diagram of the flow regulator of a liquid of another type is shown in Fig. 79a. Regulator consists of two throttle/chokes - the washer 1 fixed resistor and automatic adjusted whose resistance is determined by the position of its throttling plunger 3. This plunger is located under spring effect 5, which attempts to displace it and to increase to the right flow area (to decrease the resistance), and the pressure differential $\Delta p_1 = p_1 = p_2$ of liquid on throttle/choke 1 whose force through piston 2 attempts to displace plunger 3 to to the left and decrease the flow area (to increase the resistance of throttle/choke).

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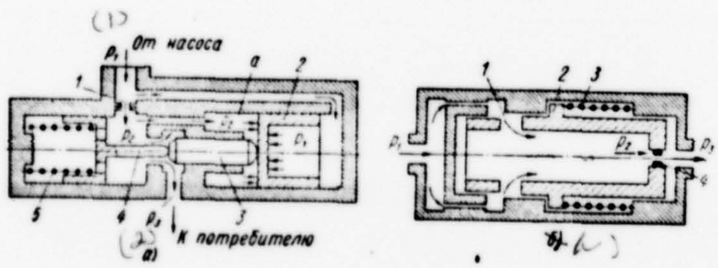

Fig. 79. Diagrams of flow regulators.

Key: (1). From pump. (2). To user.

The pressure differential $\Delta p_1 = p_1 - p_2$ is determined by the characteristic (force) of spring 5 and in practice does not depend on pressure p_1 at the inlet into regulator and pressure p_3 at output/yield from it.

During a pressure increase p_2 in the interthrottle chamber a over that which was assigned, a jump/drop in the piston pressure 2 $\Delta p_1 = p_1 - p_2$ decreases (with p_1 = const), and spring 5 through pusher 4 it displaces throttling plunger 3 to the right, increasing the section of throttle/choke. With decompression p_2, the process flow/lasts in reverse order. Consequently, the pressure differential $\Delta p_1 = p_1 - p_2$ on throttle/choke 1 is supported by constant, in accordance with which by constant will be also the flow rate through it of liquid.

The equilibrium condition of piston 2, of the manager throttling plunger 3, takes the form

$$\Delta p_1 F_0 = P \pm T - R$$

or

$$\Delta p_1 = \frac{P_0 + Cx \pm T - R}{F_0},$$

where P is compression of spring 5 with x = 0 (it corresponds to the minimum expenditure slot); T - the frictional force of moving elements; R - flow forces (reaction) of fluid flow, which acts on plunger 3; P_0 is a force of initial tension of spring (at the maximum value of x); C - the coefficient of spring constant; x - the displacement/movement of plunger 3 from completely enclosed position; F_0 is piston clearance 2.

Since the force of friction T makes the characteristic of regulator worse, they attempt to maximally decrease it. The simplest structural/design method of this is the cutting on plunger 3 and the piston of 2 circular grooves, and also the provision for accuracy/precision and finish quality.

For increase of sensitivity the controls increase area F_0 piston

2 whose value in many instances they lead to values, 5 times and more exceeding the area of plunger 3. An increase in the sensitivity is caused in this case by the fact that with an increase in the diameter of piston the perimeter of friction is raised proportional to the first degree of diameter, and area - it is proportional to its square.

Total resistance Δp (the pressure differential in regulator):

$$\Delta p = \Delta p_1 + \Delta p_2,$$

where $\Delta p_1 = p_1 - p_2$ - the pressure differential on throttle/choke 1; $\Delta p_2 = p_2 - p_3$ is the pressure differential in the slot of the adjustable throttle/choke.

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Fluid flow rates through the opening/aperture of throttle/choke 1 Q_1 and the slot (window) Q_2 of the adjustable throttle/choke are connected according to thelaw of flow continuity by expression

$$Q_1 = Q_2 = Q \quad \text{or} \quad Q = \mu_1 f_1 \sqrt{\frac{\Delta p_1 \cdot 2}{\rho}} = \mu_2 f_2 \sqrt{\frac{\Delta p_2 \cdot 2}{\rho}};$$

hence the pressure differential on the adjustable throttle/choke

$$\Delta p_2 = \frac{\mu_1^2 f_1^2}{\mu_2^2 f_2^2} \Delta p_1,$$

where μ_1 and μ_2 - the coefficients of the flow rate of the opening/aperture of throttle/choke 1 and of the slot, formed by plunger 3; f_1 and f_2 are flow passage cross-sectionals area of the opening/aperture of throttle/choke 1 and of the slot, formed by plunger 3.

The coefficients of flow rate μ_1 and μ_2 are determined by spills (on the basis of experimental data), for the precomputations it is possible to accept μ_1 = 0.62, μ_2 = 0.7-0.75.

Approximately the reaction of fluid flow to plunger 3

$$R = 2\mu_2^2 f_2 \Delta p_2 \cos \beta,$$

where β is the angle of departure of jet, depending on the form of the overlapping part of the plunger (with sharp straights-edge it is possible to accept $\beta = 69°$).

The diagram of a regulator of another type is shown in Fig. 79b. Regulator has two the consecutive hydraulic throttle/chokes, from which throttle/choke 4 is uncontrolled, definite a jump/drop in pressure $p_2 - p_3$ on piston 2, and throttle/choke 1 it is automatically correcting the fluid flow rate depending on load. The operating principle of this regulator is based on the comparison of the force, developed pressure differential $p_2 - p_3$ on piston 2, with the force of the tightening of the springs of 3 regulators, the difference between which is utilized for an automatic control for the target/purpose of the stabilization of flow rate.

Let us assume that at constant working pressure (p_1 = const) fluid flow rate was raised. In this case will be raised also a jump/drop in pressure $p_2 - p_3$, in consequence of which piston 2 under the action of this jump/drop, moving to the side of the coverage of working windows, decreases their sectional area and decreases

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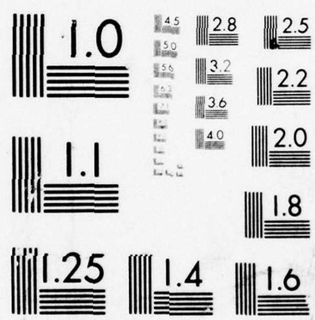

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respectively the flow rate through automatically regulated throttle 1.

During a reduction in the flow rate, occurs reverse process - piston 2 is moved to the side of discovery/opening working windows, as a result the sectional area of the latter and, consequently, also the fluid flow rate they will increase.

Synchronizers of the motion of node/units.

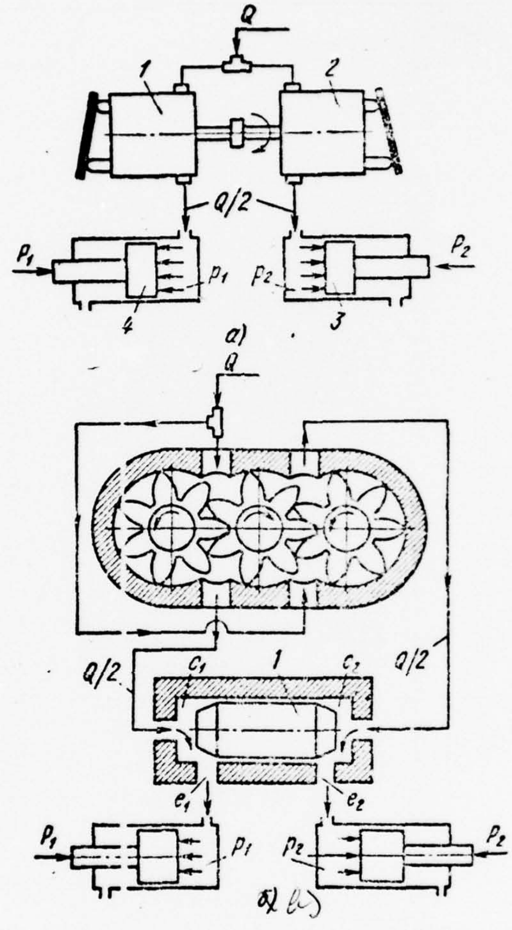
In many instances it is required to automatically synchronize the exit velocities of several hydraulic engines (actuating cylinders), which feed from one (common/general/total) pump. In this case, for the synchronization of the action of several hydraulic engines, usually it is required to ensure the equality of their velocities.

For this, are applied different equipment/devices, most widely accepted from which are equipment/devices, called the divider/denominators of the conducted/supplied flow, constructed on

volumetric or choke principle.

Volume type simplest divider/denominators are paired (connected with shafts) hydraulic motors of 1 and 2, predominantly axial-piston types (Fig. 80a). The hydraulic motors in this diagram are the flow-meter equipment/devices (dosing devices), which pass through themselves in one revolution liquid in the volume, equal to the working volume of hydraulic motor (not allowing for leakages in hydraulic motor).

Fig. 80. Diagrams of the divider/denominators of volumetric type flow.



With the equal working volumes of hydraulic motors, 1 and 2 division of flow Q of the liquid, which enters from the source of flow rate (pump) between hydraulic engines 3 and 4, will be produced in this diagram to the equal parts Q/2. Under the condition of the equality of the external load of cylinders ($P_1 \approx P_2$) the pressure differential in hydraulic motors will be caused only by friction, i.e., hydraulic motors in this case virtually will work in no-load, in view of which hydraulic slipes in them virtually are absent, i.e., volumetric efficiency them is approximately equal to unity, in consequence of which a similar synchronizing circuit under these conditions will be able to ensure relatively high accuracy/precision.

However, with the different external loads of hydraulic engines ($P_1 \neq P_2$) the equality of pressures in them will be disrupted ($p_1 \neq p_2$), as a result in the line of the underloaded hydraulic engine, it will appear margin of power, in view of which the being found on this line hydraulic motor-dosing device will enter the work as the drive of the second hydraulic motor, which is found in the line of the overloaded hydraulic engine which in this case will work in the mode/conditions of the pump, which raises pressure over the pressure of the power supply (at the inlet into hydraulic motors) of p_H to the value, necessary for the overcoming of resistance in the line of the overloaded hydraulic engine.

It is obvious that in this mode/conditions ($P_1 \neq P_2$ or $p_1 \neq p_2$) the pressure differential in both hydraulic motor-dosing devices will be caused not only by mechanical losses, but also the difference in the loads of cylinders P_1 and P_2, which in this diagram is compensated for by work as the pump of the hydraulic motor, established/installed in the branch of the overloaded cylinder.

The pressure differential on hydraulic motors, including that hydraulic motor which will work as pump, in this case there will be equal not allowing for frictional forces in system

$$\Delta p = 0{,}5 p_{num} = \frac{P_{max} - P_{min}}{F},$$

where $p_{n\,m} = \frac{P_{max} - P_{min}}{F}$ - pressure in feed line (pressure before hydraulic motors); P_{max} and $P_{\mathrm{i\,in}}$ are the maximum and minimum current load of hydraulic engines (cylinders); F - the area of cylinder.

Coefficient 0.5 is caused by the fact that the power, required for the compensation for the difference in the loads of $(P_{max} - P_{min})$, is distributed equally between both hydraulic motors.

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Consequently, with no load of one of the cylinders both hydraulic motor-dosing devices will work (not allowing for losses for friction in system) with the pressure differential, equal to 0.5 p_{num}, where the p_{num} - pressure in feed line, which corresponds to the load of P_{max}.

In view of the possibility of disturbance/breakdown because of this, the accuracy/precisions of dosing with the varying loads of hydraulic engines as dividers can be applied in the examined simple diagram only the hydraulic motors, which possess high volumetric efficiency (small leakages), to such are related the hydraulic motors of axial-piston types.

Figure 80b gives the diagram of the volumetric divider/denominator of flow, in which as flow-meter equipment/device is applied three-gear pump, which is actually two combined gear hydraulic motors. For a decrease in the possible error of the division of flow, caused by a difference in loads P_1 and P_2 actuating cylinders, is applied the automatic throttle governor, which is floating plunger 1, which at the equal pressures p_1 and p_2 liquid in the lines, which drive to hydraulic engines ($p_1 = p_2$), will be found in the mid-position between channels e_1 and e_2 through which occurs the power supply of these engines. However, during load change in one of the engines ($P_1 \neq P_2$) plunger 1 as a result of the created pressure difference ($p_1 \neq p_2$) of liquid in chambers c_1 and C_2 it will be displaced in the direction of the chamber with less pressure and partially it will overlap the appropriate channel of the power supply of engines e_1 and e_2, in consequence of which the total resistance (and, consequently, fluid flow rates) of the branches of both engines will be made even.

It is obvious that not allowing for friction plunger 1 with any, as small as desired disturbance/breakdown of equality $p_1 = p_2$ comes

into action compensating possible mismatching whimight come about as a result of the change of leakage in the hydraulc motors with $p_1 \neq p_2$. Consequently the hydraulic motor-dosers in this diagram will operate at a constant pressure differential caused only by frictional losses in them, thanks to which equality of flow ofthe liquid in them will be ensured.

Choke divider/denominators of flow. The divider/denominator of flcw (valve of the relationship of flows) is intended for the separation of one flow of working fluid to two or more flows.

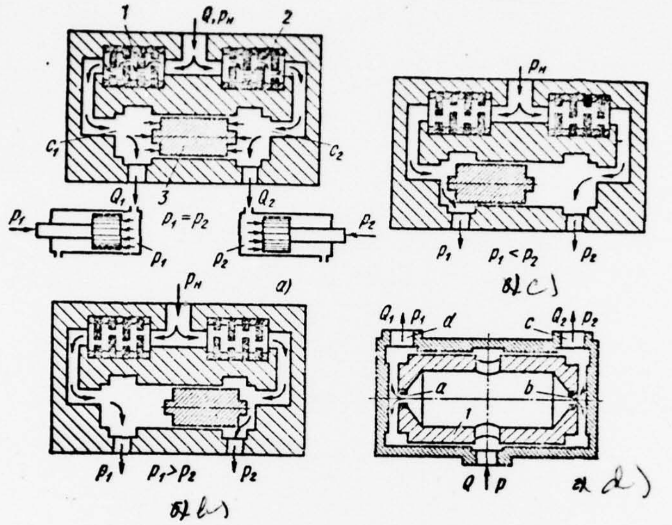
From choke dividers most is widely common the equipment/device whose diagram is shown in Fig. 81a.

The division of flow Q in this equipment/device on Q_1 and Q_2 is realized with the aid of two packages of the chcke flanges of 1 and 2 and examined floating plunger 3 of adjustable throttle/choke, which automatically ensures the equality of pressures in chambers c_1 and C_2, connected with the cavities of hydraulic engine.

Plunger 3 as in the diagram examined above (see Fig. 80b), it will be located at the equal loads $P_1 = P_2$ hydraulic engines and the respectively equal pressures of liquid in chambers c_1 and C_2 in free position (between the cutoff edges of these chambers through which is realized the power supply of these engines). With the disturbance/breakdown of the equality of the load of engines ($P_1 \neq P_2$ and respectively $p_1 \neq p_2$) plunger 3 as a result of the created pressure difference in chambers c_1 and C_2 will move in the direction of the chamber with less pressure and partially will overlap the appropriate channel of the power supply of engine, in consequence of which the total resistance (and, consequently, fluid flow rates) of the branches of both engines they will be made even.

Figure 81b depicts the position of the plunger when $p_1 > p_2$ and respectively $P_{c1} > P_{c2}$, while Fig. 81b - the situation, when $p_1 < p_2$ and respectively $p_{c1} < p_{c2}$.

Fig. 81. Diagrams of the divider/denominators of choke type flow.



Since the system is supplied from common source (pump) with the pressure of $p_н$, the force condition of equilibrium, which act on choke plunger 3, will have the form (as engines are applied actuating cylinders)

$$\Delta p_{11} + \Delta p_{31} + p_1 = \Delta p_{22} + \Delta p_{32} + p_2 = p_н,$$

where of the $\Delta p_{11} = p_н - p_{c1}$ and $\Delta p_{22} = p_н - p_{c2}$ - loss of pressure in lamellar throttle/chokes 1 and 2 (in accordance with given $\Delta p_{11} + p_{c1} = \Delta p_{22} + p_{c2}$); $\Delta p_{31} = p_{c1} - p_1$ and $\Delta p_{32} = p_{c2} - p_2$ - the loss of pressure, caused by the partial overlap by the plunger of 3 channels of the power supply of the hydraulic engines of cylinders; here $p_{c1} = p_1 + \Delta p_{31}$ and $p_{c2} = p_2 + \Delta p_{32}$ - pressure in chambers c_1 and C_2; p_1 and p_2 are pressures in cylinders.

Disregarding losses due to friction of plunger 3, we will have an $p_{c1} = p_{c2}$, in accordance with which

$$p_1 + \Delta p_{31} = p_2 + \Delta p_{32}.$$

In such a case, when the piston stroke of one of the cylinders along some reason ceases itself, plunger 3 completely will overlap the window of the power supply of paired cylinder, as a result the motion of its piston also will cease itself.

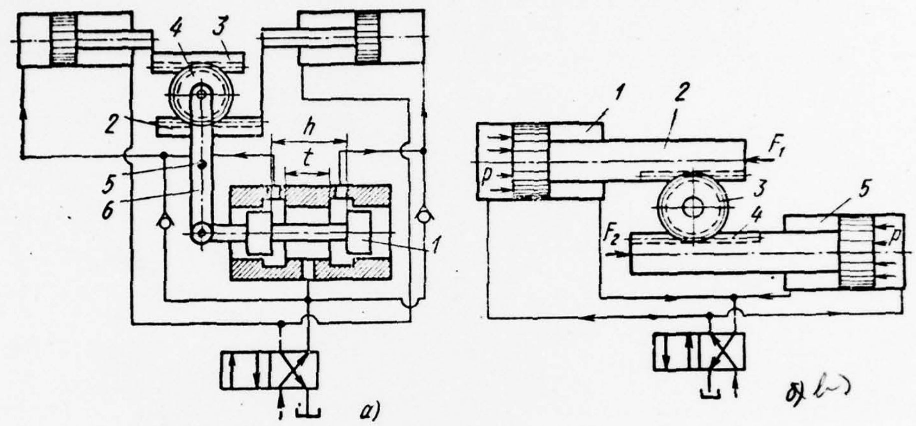
From that which was given it follows that under the taken condition of $p_{c1} = p_{c2}$, $\Delta p_{11} = \Delta p_{22}$ and $F_1 = F_2$ any by a change in their load P_1 and P_2, will be accompanied by an equal, but opposite on sign change in the adjustable resistor (jump/drop Δp_{31} and Δp_{32}), reached to the displacement of choke plunger 3.

In actuality, as a result of friction of plunger occurs certain

error in the division according to flow rate, whereupon during a decrease in the flow rate the relative error of division grow/rises and during some flow rates, friction of plunger 3 and resistance of the package of choke flanges divider/denominator will not react to a change in the flow rate. Taking into account this, one should maximally lower the frictional forces of plunger and raise within margins the resistance of constant throttle/chokes. The error of the division of the flow (difference in flow rates Q_1 and Q_2 in the branches of actuating cylinders, referred to the complete flow rate of Q) of these divider/denominators does not exceed 2-3o/o.

For the division of fluid flow to two, assigned parts apply also the divider/denominators of flow (sample preparing devices) with one movable cell/element.

Fig. 82. Synchronizing circuits of the engines: a) hydromechanical and b) mechanically.



This divider/denominator consists of two in parallel established/installed hydraulic friction a and b (Fig. 81d), the accomplished/carried out in the form throttle/chokes in the bases of floating piston 1. Applied in volume Q liquid is divided with the aid of these friction into two flows Q_1 and $1Q_2$ (during equal friction these flows are equal).

If the flow rate in the main line, connected, for example, with channel d, will exceed for any reason the flow rate in main line c, it will be created the difference in the friction of throttle/chokes; loss of pressure on throttle a exceeds loss of pressure on throttle/choke b, as a result pressure p_2 will exceed pressure p_1. Under the action of the unbalanced force, caused by the created pressure differential ($\Delta p = p_2 - p_1$), floating piston 1 will move into the position in which equality $p_2 = p_1$ will be restore/reduced, thanks to which will be ensured equality $Q_2 = Q_2$.

Slave/servo type synchronizers. More advanceed is the synchronization with the aid of the hydraulic slave/servo

differential attachment, which consists of two connected with the stock/rods actuating cylinders of racks 3 and 2 and of gear 4, connected through lever 6 with the plunger of distribution valve 1 (Fig. 82a).

The operating principle of synchronizer is based on the automatic resistance control of the expenditure windows of distribution valve in the function of the load of actuating cylinders, realized by the indicated follower.

For providing a synchronism of motion, the distributor must be executed so that in the mid-position of plunger would be provided for necessary negative overlap ($t > h$) and plunger it was arrange/located symmetrically relative to the windows of the power supply of the right cavities of actuating cylinders.

From diagram it follows that during the synchronous piston stroke of cylinders gear 4 will be turned around its axle/axis in the constant/invariable position of the latter. During the disagreement/mismatch of the velocities of cylinders, caused by the changes in the load or by other reasons, the gear will be

additionally be moved (rolled) in the rack of the delaying cylinder, as a result lever 6 will turn itself around axle/axis 5 and will move the plunger of valve 1 into the appropriate side, increasing the friction of the feed line of the liquid of leading cylinder and decreasing the friction of the delaying cylinder.

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As a result of the proceeding in this case redistribution of the fluid flow rates into cylinders, the synchronism will be restore/reduced.

A similar diagram is not sensitive to volumetric losses and changes in the load of cylinders and friction, or it differs in terms of the high accuracy/precision of synchronization in displacement/movement and piston speed of cylinders.

Synchronization of hydraulic engines by mechanical communication/connection. For synchronization of the rates of oncoming movement of the two power cylinders 1 and 5 is used also the

diagram with supplementary mechanical connection of their stock/rods through gear 3, racks 2 and 4, connected with the pistons of the corresponding cylinders (Fig. 82b). This diagram corresponds to the case of the parallel arrangement/permutation of friction. During the matched motion of the stock/rods, gear 3 will be turned around its fixed axis without load. During the disagreement/mismatch of speeds, caused by change, for example, the loads of any out of cylinders, part of the thrust, developed with the underloaded (anticipate/loading) cylinder, will be transferred through the gear 3 and corresponding racks (2 or 4) on the stock/rod of the delaying cylinder, being added to the force, developed with the piston of this cylinder.

Gear 3 in this diagram is the mechanical compensator of the difference in the loads of cylinders, and its communication/connection with racks replaces the adjustable resistor of the examined floating plunger.

From the given diagram it follows

$$(p_1 f_1 - F_1) - (p_2 f_2 - F_2) = \frac{M}{r},$$

where p_1, f_1, p_2 and f_2 - respectively pressure in cylinders and their area; M and r it follows torque/moment on gear and its radius.

Under condition $p_1 = p_2$ and $f_1 = f_2$, we have

$$F_2 - F_1 = \frac{M}{r}$$

or

$$M = (F_2 - F_1)\, r.$$

Consequently, the transferred through this rack- gear mechanism calculated torque/moment is caused by the difference in the loads of cylinders. In accordance with this, the rack-and-pinion gear must be designed for strength taking into account the forces, caused by the possible disagreement/mismatch of the load of cylinders. For example during the possible (assigned) 25o/o- nominal load variation rack mechanism must be designed for force 0.25F, where $F = p\omega$ are designed the calculated nominal force of cylinder by area ω.

For the synchronization of the motion of two cylinders, which move in one direction, must be applied supplementary (idle) gear.

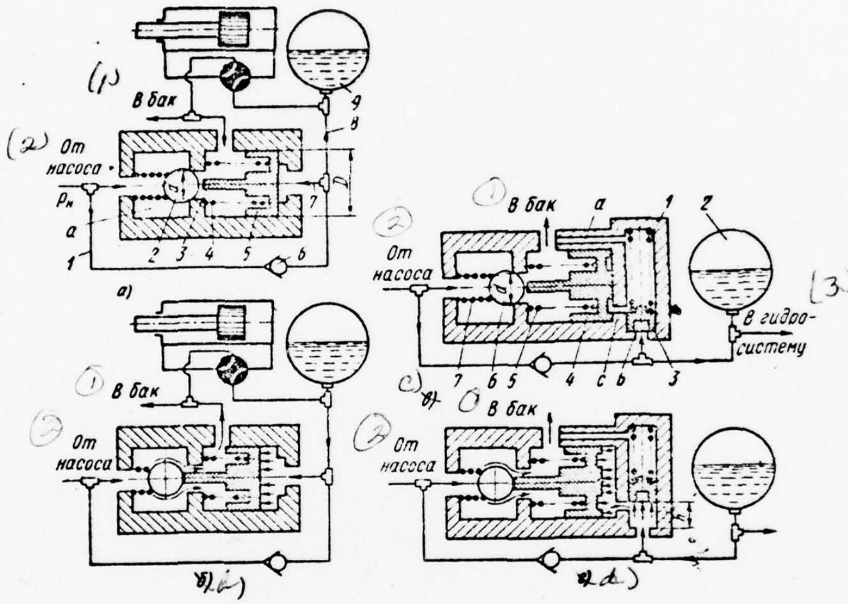
Equipment/devices for the automatic discharging of the uncontrolled pumps.

In the hydraulic systems with the occasional consumption of liquid, equipped with the pump of constant feed, are applied the equipment/devices (automatic machines of discharging) for the automatic discharging of pump by means of the translation/conversion of pump upon achieving the assigned pressure into the mode/conditions of idling.

The application/use of these automatic machines provides for the presence in the hydraulic system of the gas-hydraulic storage battery/accumulator (see page 131), which in the operating cycles of pump under the conditions of idling compensates for (it completes) hydraulic slip in hydraulic system, and also it supplies by liquid users with small flow rate. After the pressure in storage battery/accumulator as a result of the consumption of liquid is lowered to the established/installed minimum value, automatic machine again switches on pump for the recharging of storage battery/accumulator, after disconnecting it from tank.

Fig. 83. Diagrams of equipment/devices for the automatic discharging of pump.

Key: (1). In tank. (2). From pump. (3). In hydraulic system.



The diagram of a similar action of direct automatic machine is represented in Fig. 83a. Liquid from pump enters chamber a and on conduit/manifold 1 through check valve 6 into conduit/manifold (channel) 8 proceeds to storage battery/accumulator 9. Then through conduit/manifold 7 it proceeds to piston 5, spring-loaded 4.

With a pressure increase in storage battery/accumulator 9 to the rated value at which is adjusted spring 4, piston 5 is moved to the left (Fig. 83b) and by its push rod 3 (see Fig. 83a) it displaces ball bearing check valve 2 (detaching it away from seat/socket). In this case, the pump will be connected with the channel, connected with tank, and consequently pressure at output/yield from pump (forcing) will be lowered virtually to zero. Storage battery/accumulator in this case is detached from tank with the aid of check valve 6.

piston 5 and valve 2 they will be located in the position, which corresponds to the work of pump in the mode/conditions of zero pressure, thus far pressures in storage battery/accumulator as a result of the consumption of liquid is not lowered to the value at which spring 4 will be able to move piston 5 to the right; ball bearing check valve 2 in this case will sit down into its seat/socket

and will cut off the line of pump from the line of tank.

The calculation of this automatic machine is made through the following diagram (by friction of movable parts and hydraulic friction we disregard). To the force of the pressure of liquid on piston 5 in the work of pump in the mode/conditions of battery charging (before discovery/opening check valve 2) counteracts the force of spring 4 and the force of pressure of liquid on valve 2

$$p_{a\kappa}F = p_{\kappa}f + P_{np} = \frac{\pi}{4}d^2 p_{\kappa} + P_{np}$$

or

$$P_{np} = p_{a\kappa}F - p_{\kappa}f = p_{a\kappa}\frac{\pi D^2}{4} - p_{\kappa}\frac{\pi d^2}{4} = \frac{\pi}{4}(p_{a\kappa}D^2 - p_{\kappa}d^2),$$

where $p_{a\kappa}$ - the pressure of liquid in storage battery/accumulator; F = $\pi D_2/4$ - piston clearance; p_{κ} - the

pressure of liquid in pump; $f = \pi d^2/4$ - the area of the passage opening, overlapped by valve; P_{np} - the force of the precompression of spring.

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The pressure liquids at which valve 2 will be unearthed,

$$p_{a\kappa} = p_{\kappa} \geqslant \frac{P_{np}}{F - f}$$

or

$$p_{a\kappa} = p_{\kappa} \geqslant \frac{4P_{np}}{\pi(D^2 - d^2)},$$

where D and d will be unearthed the diameters of piston and opening/aperture, overlapped by ball/sphere.

After the opening of check valve, 2 pressures hearth and above it are equalized, i.e., $p_{н}f \approx 0$, and consequently, equilibrium condition will take the form of $Fp_{max} = P_{np}$. In accordance with this the pressure of liquid in the storage battery/accumulator with which the valve will be closed,

$$p'_{ак} \geqslant \frac{P_{np}}{F}$$

or

$$p'_{ак} \geqslant \frac{4P_{np}}{\pi D^2}.$$

By the corresponding selection of diameters D and d and of the force of the spring of aaaaa it is possible to obtain the required interval of pressures, which determines torque/moments it began openings and the closures of check valve 2.

With the large feeds also of pressures, that determines torque/moments it began openings and the closures of check valve 2.

In large feeds and pressures, are applied two-stage relief mechanisms (automatic machines). In these equipment/devices (Fig. 83c) storage battery/accumulator 2 connect with the piston chamber b of pilot valve 3 of first stage. The small piston of this valve upon reaching of the maximum pressure compresses spring 1, it steps down (Fig. 83d) and it connects the line of storage battery/accumulator with channel c, which drive into the cavity of dump piston 4 (Fig. 83c). The latter under the effect of pressure of liquid is moved to the left and its dowel it open/discloses ball bearing check valve 6, after connecting the line of pump with tank.

Because of the fact that past rotary-plunger valve 3 passes small amount of liquid, required for the lift of dump piston 4, the

diameter of the seat/socket of this valve (and the force of spring 1) usually moderate (3-5 mm).

The pressures at which occurs opening $(p_{a\kappa})$ and of the $(p'_{a\kappa})$ of relief valve, depend on the stiffness coefficient of spring 1 and of the degree of its compression within the limits of the displacement of valve 3 to value h from the downhand position in which the cavity of dump piston 4 is connected through channel a with tank (Fig. 83d), at the overhead position, in which this cavity is connected with storage battery/accumulator (see Fig. 83c).

The pressure of the $p'_{a\kappa}$ of the power supply of the storage battery/accumulator with which piston 4 is located in downhand position (check valve 6 is enclosed), will be determined by the precompression of spring 1 (pressure in drainage line we disregard):

$$P_{np\,\min} = fp'_{a\kappa} \leqslant Cx;\ p'_{a\kappa} = \frac{Cx}{f},$$

where $P_{np\,\min}$ we disregard the minimum force of compression of

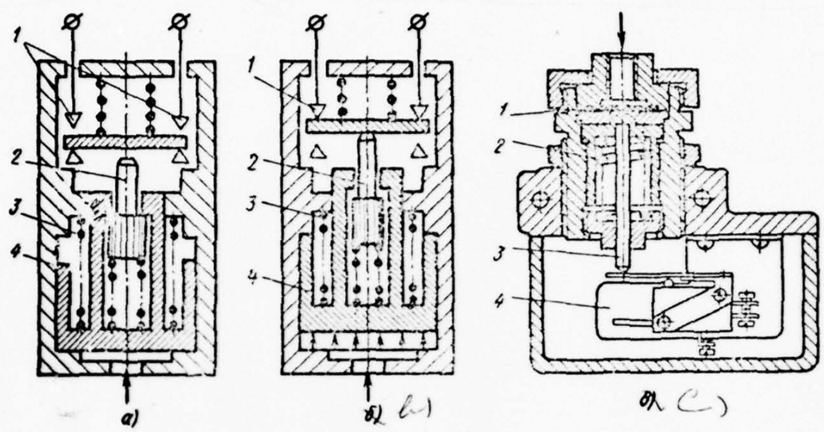
spring 1.

The pressure of $p_{aк}$, which corresponds to the torque/moment of the translation/conversion of pump into the mode/conditions of idling (in this case channel c is connected with storage battery/accumulator),

$$P_{np\,max} = fp_{aк} = Cx + Ch = C(x + h);$$
$$p_{aк} = \frac{C\,(x + h)}{f},$$

where $P_{np\,max}$ is connected the maximum force of compression of spring 1; C and x are a stiffness coefficient of spring and its preliminary (assembling) compression, which corresponds to the downhand position of valve 3; h - supplementary compression of spring 1 during valve lift 3 at the position by which channel c it will be connected with storage battery/accumulator.

Fig. 84. Electrohydraulic disconnection switches (pressure relay.



In accordance with this

$$\frac{p'_{ак}}{p_{ак}} = \frac{x}{x+h}.$$

Effort/force P_5 spring 5 depends on the maximum pressure of $p_{ак}$ in storage battery/accumulator (pressure cf gutter we disregard), the piston clearance 4, of force P_7 spring 7, which presses valve 6 to saddle, and the area of its saddle f with a diameter of d:

$$P_5 = p_{ак}f_4 - P_7 - p_{ак}f_6.$$

The minimum effort/force of this spring must be so that would be provided for the overcoming of the frictional forces of piston 4.

Piston stroke 4 must be sufficient for providing required valve opening 6.

Pressure relay.

Pressure relay is applied in electrohydraulic automatic control for the transmissions of the control signals at a distance. Momentum/impulse/pulses for the function of relay serves a pressure increase in its circuit. This pressure is converted into the rectilinear or angular displacements of spring-loaded of plunger or diaphragm/membrane, as a result of which they are closed or they are broken electric contacts depending on their designation/purpose.

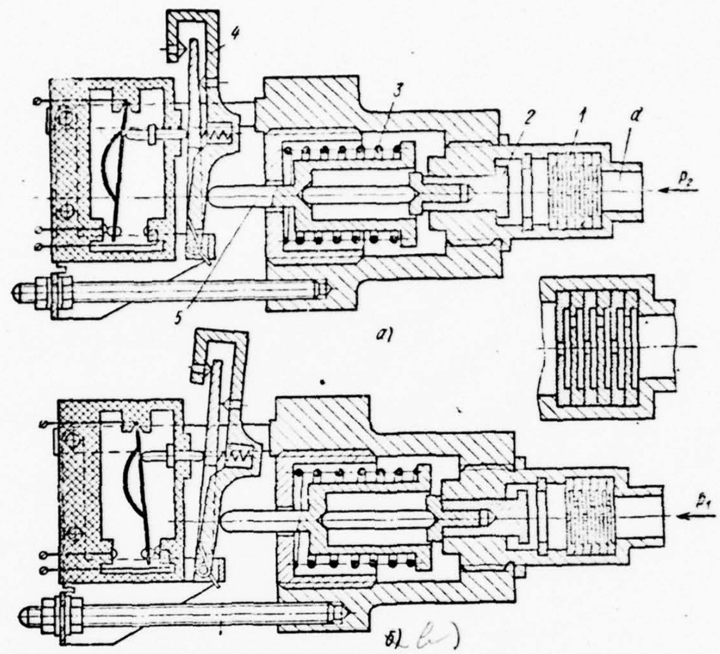
Simplest of these relay are the electrohydraulic disconnection switches (Fig. 84), designation/purpose of which is the closing/shorting and breaking of signal electrical circuit. Figure 84a disconnection switch depicts in the situation, when piston 4

under the action of the effort/force of spring 3 is drowned and electric contacts 1 extended. During applied pressure increase to the value by which is overcomed the effort/force of spring 3, the latter is compressed, and piston 4 through spring-opposed pusher 2 closes electric contacts 1 (Fig. 84b).

For providing an airtightness, is applied by the pressure relay of membrane/diaphragm type (Fig. 84c). The pressure of liquid acts on spring-loaded 2 flexible diaphragm/membranes 1, with sagging/deflection of which is put into action through the pusher 3 electric switch 4 control circuits.

Similar relays are discharged for a pressure to 200 kgf/cm² and above. The insensitivity of relay (the pressure differential of connection/inclusion and the disconnection) depends on operating pressure and usually at pressure 200 kgf/cm² does not exceed 10 kgf/cm².

Fig. 85. Pressure relays of piston type.



Is applied also the pressure relay of piston types, whereupon them frequently they combine with safety valve. Figure 85a gives schematic of one of the piston relay. Liquid under pressure proceeds to channel a and, in passing by the throttle, which consists of the package of washers 1, it proceeds to spring-loaded to plunger 2. During a pressure increase to

$$p_2 > \frac{P}{f},$$

where f - the area of plunger in which is overcomed effort/force P of spring 3 (friction of plunger we disregard), plunger is moved to the left and through pusher 5 and electric switch 4 closes the circuit of the corresponding contacts.

During a decompression to

$$p_1 < \frac{P}{f}$$

spring 3 wrings out piston 2 to the right, as a result the electric switch closes the circuit of other contacts (Fig. 85b).

Timing relay.

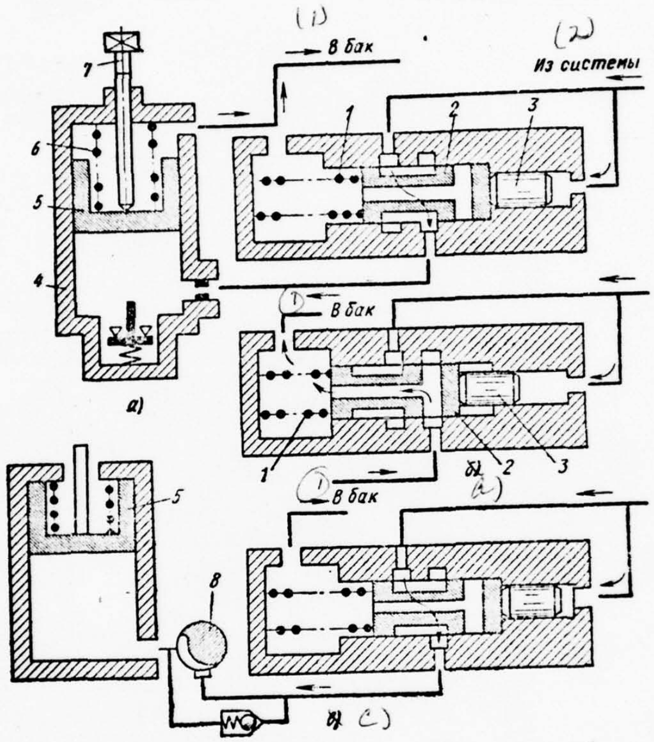
In many hydraulic systems it finds relaying (valve) of delay.

Valve of delay (timing relay) - the distributive apparatus, intended for the connection/inclusion of the flow of the working fluid through the determined time interval after the achievement of the established/installed pressure in the supplying hydraulic lines. With the aid of this relay are realize/accomplished the adjustable delay between by two following each other phases of the motion of performing aggregate/units or the controlled delay of certain time

interval of any signal.

Fig. 86. Schematics of the timing relay.

Key: (1). In tank. (2). From system.



The assigned time interval of the holding time of relay is determined by opening time by the liquid of the special cylinder (tank) whose piston in end position affects the appropriate valve (or to the disconnection switch of the circuit of solenoid tap/crane), or by time of the escape of liquid from cylinder (by time of the overflowing through the throttle/choke of liquid of one tank into another).

The schematic of the timing relay, in which the delay is determined by time of the displacement by the piston of liquid from cylinder during alternating/variable piston stroke and fixed resistor, is depicted on Fig. 86a. The position of plunger 2 relays here corresponds to execution by the hydraulic engine of working operation. In this position the cavity of cylinder with 4 relays is connected through plunger 2 with the working line of hydraulic system, and piston 5 it steps down to fixing screw 7, which limits its course.

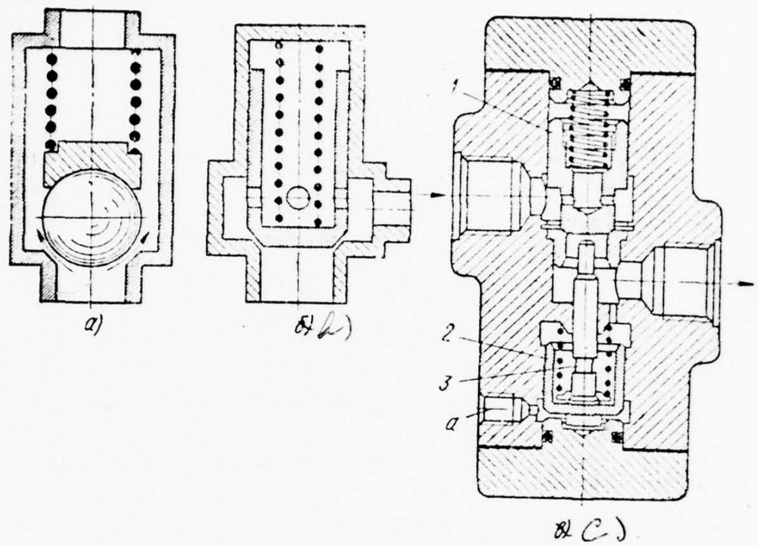
At the termination of working operation, the pressure in the working line of hydraulic system is raised, as a result plunger 2 under the effect of pressure of liquid on plunger 3 will move, after overcoming the effort/force of spring 1, to the left (Fig. 86b) and

will connect the cavity of cylinder 4 with tank. After this piston 5 under spring effect 6 will move down, displacing liquid into tank.

Time of the displacement of piston 5 of the overhead position in lower changes with the control of the length of screw/propeller 7 and with the resistance, established/installed on output/yield from cylinder 4. At the end of the piston stroke, 5, affecting the limit switch (or to the valve of changeover), realize/accomplishes a disconnection or a reverse of system.

Figure 86c shows the schematic of relay, in which the holding time is determined by the control of throttle/choke 8 with constant piston stroke 5.

Fig. 87. Check (locking) valves: a and b) unguided; c) controlled.



Locking (reverse/inverse) valves.

Reverse/inverse valve fulfills in hydraulic system the same functions, as the rectifier in electrical circuit, passes in the absence of the extraneous control pressure fluid flow only in one direction. In the presence of this effect, the valve passes flow in both directions.

Check valve (Fig. 87) is structurally similar protective (see Fig. 62), with by that only difference, that in it are applied the uncontrolled springs with a small effort/force, sufficient for reliable landing/fitting gate in seat/socket.

Are common valves with ball bearing (Fig. 87a) and conical (Fig. 87b) gates.

Valves with conical gate possess more high speed operation, than ball bearing. The triggering time of valves varies depending on size/dimensions and the construction from 0.1 to 10 ms.

In automatic hydraulic equipment are applied also the controlled check valves (with governing external agency). The valve of this type (Fig. 87c) ensures free passage of liquid in one direction, in opposite direction passage is ensured by positive opening gate 1 with the aid of the pusher of 2 pistons 3. For this, the liquid from governing apparatus is supplied through channel a under piston 3.

Hydraulic locks.

Hydraulic lock is the distributive hydraulc apparatus, intended for the automatic closing of liquid in the cavities of hydraulic engine for the target/purpose of the fixing of the piston of actuating cylinder in the assigned positions. Schematic diagram of one of them is depicted on Fig. 88a. In the housing of lock 1, are placed two reverse/inverse (locking) ball valve 2 and 4, between which is placed floating small piston 3. Liquid from distributor proceeds to the lock through channels a and b and from lock actuating cylinder 5 through channels c and d.

During the supply of liquid to channel a (Fig. 88a) blows away left check valve 2, and the liquid past channel d passes into the left cavity of actuating cylinder 5. In this case, by the pressure of liquid small piston 3 is displaced and breaks away to the right and opens the right cut-off valve 4, ensuring the passage of the liquid, abstract/removed from channel c, connected with the nonoperative (left) cavity of actuating cylinder 5, into channel b and further to distributor.

Fig. 88. Schematic of hydraulic lock.

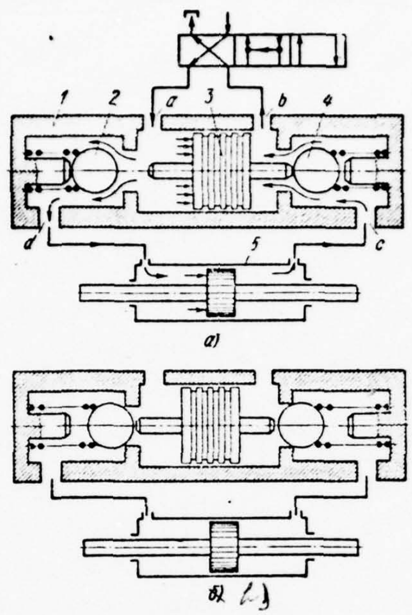
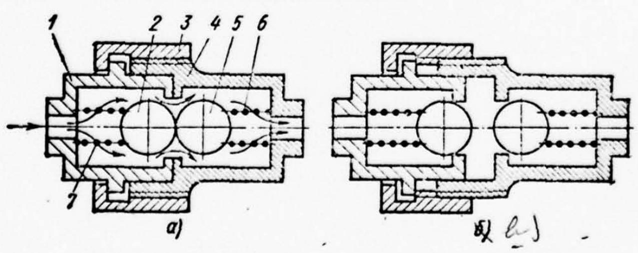


Fig. 89. Disconnectable coupling.



During the supplying of liquid from distributor into channel b (with reverse) lock wear/operates analogously, but in opposite direction.

In such a case, when the circulations of liquid does not occur (which corresponds to the mid-position of distributor in Fig. 88b), check valves 2 and 4 cut off liquid in the cavities of actuating cylinder 5, fixing its piston and holding its load in the assigned position.

Disconnectable coupling.

In the hydraulic apparatuses which undergo frequent disassembly, usually are applied the equipment/devices, which prevent in this case the discharge of liquid. As such equipment/devices serve the special disconnectable coupling in which with the dissociation of conduit/manifold cut-off valves automatically overlap flow areas.

Schematic of the simplest disconnectable clutch with the ball bearing gates (valves) of represented in Fig. 89. In the installed

form (Fig. 89a) of part 1 and 4, connected with the ends of the diconnected conduit/manifolds, are tightened by adapter nut 3. In this case, ball bearing gates 2 and 5 come into contact friend with other and, wringing out from their saddles, is formed passage for a liquid.

With the screwing of adapter nut, parts 1 and 4 will move away from each other, and ball bearings they are set under the effort/force of springs 6 and 7 into their saddles, airtightly overlapping conduit/manifolds (Fig. 89b).

Valves of the start of emergency system.

In many instances is required to ensure the duplicating (emergency) supply of hydraulic engine with the failure of the system of the basic supply.

Figure 90a and b shows the schematic of the switch (shuttle valve) of a similar designation/purpose with the being broken joint. Switch is intended for automatic connection of user to basic a or

doubling b to hydraulic system upon starting of one of them. Figure 90a shows the work of the basic system, Fig. 90b - the work of the duplicating system.

Fig. 90. Switch to the emergency service of hydraulic system.

Key: (1). Abstract/removed pressure.

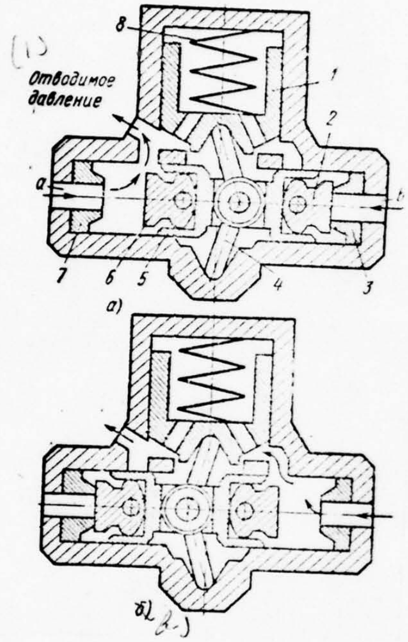
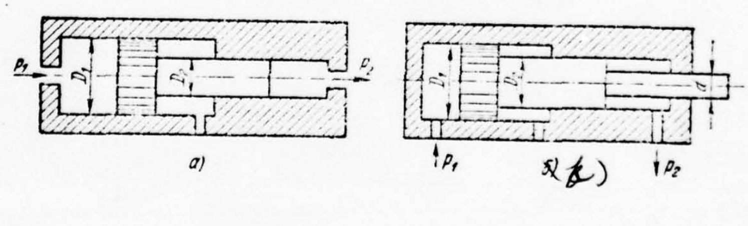


Fig. 91. Direct-acting pressure transducers of single action.



With damage of one of the conduit/manifolds, changes the pressure on the end/faces of shuttle valve, which causes its displacement, whereupon the supply of user is realize/accomplished through the second duplicating conduit/manifold.

For the fixing of the position of shuttle and elimination of false response, are applied different fixers. In the schematic of the switch, presented in Fig. 90, the fixing of piston 5 is realize/accomplished with the aid of being broken joint 4, loaded through piston 1 by spring 8. The kinematics of this joint is such, which for the displacement of the piston 5, of the carrier flat/plane rocking valves 2 and 6, of one position into opposite will be required to move upward piston 1, what impedes spring 8. This spring presses also through joint 4 gate 2 to appropriate seat/socket 7 or 3 housings, ensuring the preliminary sealing/pressurization of valve, which after will appear the pressure of liquid, is raised because of the unbalanced effort/force of this pressure on piston 5.

HYDRAULIC VOLUMETRIC CONVERTERS.

In the practice of the application/use of hydraulic drive,

frequently appears the necessity for the equipment/devices, which convert pressures or flow rates. Similar equipment/devices were called the name hydraulic converters.

In the general case hydraulic transformer is called the volumetric hydraulic machine, intended for the energy conversion of one flow of working fluid into the energy of flow with other values of pressure and flow rate. In this case, they distinguish:

the direct-acting hydraulic transformer, comprised of two hydraulic cylinders of the different diameters whose pistons are strictly connected between themselves;

the rotary hydraulic transformer, comprised of hydraulic motor and pump with the different working volumes whose shafts it is strict the connection between themselves.

The schematic diagram of the direct-acting hydraulic transformer of the single action, which raises pressure, is given in Fig. 91a. Pressure p_1 applied liquid acts on piston clearance

$$F_1 = \frac{\pi D_1^2}{4};$$

exit pressure it acts on stock/rod's less area

$$F_2 = \frac{\pi D_2^2}{4}.$$

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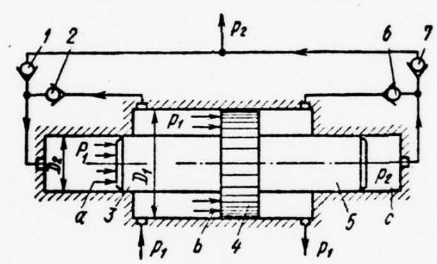

Fig. 92. Direct-acting pressure transducer of double action.

The factor of amplification of the pressure hearth by which understands the ratio of cutlet pressure to input, is equal (neglecting friction)

$$i=\frac{F_1}{F_2}=\frac{D_1^2}{D_2^2}=\frac{p_2}{p_1},$$

where F_1 and D_1 are an area and the diameter of the section of piston (cylinder) F_2 and D_2 is an area and the diameter of stock/rod's section; p_1 and p_2 is an input and outlet pressure of liquid.

Figure 91b gives the schematic of the converter with differential piston with the aid of which it is possible to obtain the required for high pressures small effective area during the simultaneous provision for hardness and structural strength of converter. In this case

$$i=\frac{D_1^2}{D_2^2-d^2}=\frac{p_2}{p_1},$$

where d is a diameter of stock/rod's shank.

For the elimination of the idling which have in the hydraulic transformers of single action, are applied the hydraulic transformers of dual (continuous) action (Fig. 92).

The supply of low-pressure cavities p_1 is realize/accomplished by the distribution valve (Fig. 92 shows), given with the piston of converter at the end of its each of courses.

During the supplying of liquid under pressure p_1 the left cavity b of cylinder, it simultaneously will enter through check valve 2 the left plunger cavity a, as a result piston 4 will be moved to the right under the action of this pressure of liquid both on the piston and on plunger 3, developing effort/force

$$P_1 \doteq \frac{\pi(D_1^2-D_2^2)}{4}p_1+\frac{\pi D_2^2}{4}p_1=\frac{\pi D_1^2}{4}p_1.$$

. Liquid under the high pressure p_2 will be displaced with piston stroke 4 to the right by plunger (stock/rod) 5 through check valve 7 and to user. Check valve 6 overlaps in this case its passage into the right cavity of cylinder, which in this case is connected through the distributor with the line of gutter, while check valve 1 overlaps its passage in the left piston and rod cavity of cylinder.

Amplification factor will be determined from equality

$$\frac{\pi D_1^2}{4} p_1 = \frac{\pi D_2^2}{4} p_2; \; i = \frac{p_2}{p_1} = \frac{D_1^2}{D_2^2} \quad \text{and}$$

$$p_2 = p_1 \frac{D_1^2}{D_2^2}.$$

During the supplying of liquid under pressure p_1 into the right cavity b of cylinder, the process will flow/last in reverse order.

Converters of this type are manufactured on productivity to 110 l/min, coefficient i of the intensification of pressure transducer it

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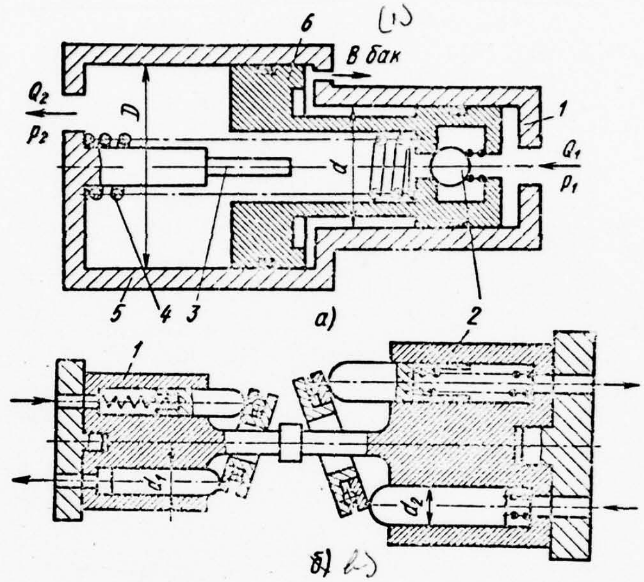
is located from 3:1 to 100:1.

Fig. 93. Schematics of the converters: a) reducing pressure; b) rotary action.



Converters, which reduce pressure.

Apply also the converters, which reduce pressure and raise under otherwise equal conditions fluid flow rate.

Their application/use from the viewpoint of energy losses frequently is more rational than the application/use of the reducing reducer (see p. 103). The schematic of a similar converter is represented in Fig. 93a. Converter consists of two twin cylinders 1 and 5 different sections with common/general/total differential piston 6. Liquid under high pressure will be fed from hydraulic system into the cylinder of 1 small section and is displaced into the main line of the user of low-pressure liquid of the cylinder of 5 large cross section.

For the return of pistons at starting position after the cessation of the supply of cylinder 1 (with p = 0) is applied spring 4.

The reduction ratio in the pressure (reduction) is determined of

relationship/ratio (friction and force spring 4 we disregard)

$$i = \frac{d^2}{D^2} = \frac{p_2}{p_1};\ p_2 = p_1 \frac{d^2}{D^2}, \qquad (47)$$

where d and D - the diameter of the cylinders of small and large cross sections.

Respectively volumetric fluid flow rates Q_2, displaced of the cylinder of 5 large cross section, and Q_1, applied into the cylinder of 1 small section,

$$\frac{Q_1}{Q_2} = \frac{d^2}{D^2};\ Q_2 = Q_1 \frac{D^2}{d^2}.$$

For the compensation for the possible hydraulic slip from the

closed main line, connected with cavity of cylinder 5 large cross section, in the schematic in question is applied ball valve 2, which at the end of the stroke of piston to the left is wrung out by dowel 3 from its saddle and opens the passage of liquid from high-pressure main line. p_1 in the quantity necessary for the completion of leakages.

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However, as soon as pressure p_2 in the cavity of the cylinder of 5 large cross section in this case will be raised to the value, which exceeds computed value, the equilibrium condition of piston [see expression (47)] will be disrupted, and it will be displaced to the right on size/dimension, that makes it possible for valve 2 to sit down into its seat/socket.

Rotary converters.

Found use also the rotary pressure transducers and flow rate.

Figure 93b shows the schematic of the rotary axially-piston converter (hydraulic converter), which is of two connected with shafts machines 1 and 2 different working volumes. When using machine 1 as hydraulic motor the calculated feed (leakages we disregard) of pump 2 will exceed the expenditure of hydraulic motor in

$$i=\frac{n_2q_2}{n_1q_1}=\frac{Q_2}{Q_1}\ \text{pa3} \quad \text{or} \quad Q_2=iQ_1,$$

where $n_2 = n_1 = n$ - the frequency of the rotation of the shafts of hydraulic machines; $q_2 = \pi d^2{}_2/4\ z_2h_2$ and $q_1 = \pi d^2{}_1/4\ z_1h_1$ - the working volumes of machine 2 (pump) and machine 1 (hydraulic motor); d_1 and D_2; h_1 and h_2; z_1 and z_2 - the diameters, power stroke and the number of pistons respectively of machines 1 and 2; Q_2 and Q_1 - the calculated feeds of machines.

The design pressure of machine 2 (pump) under the condition of the equality of the power of machines lower than the pressure p_1 the supply of machine 1 (hydraulic motor) is determined by relationship/ratio (mechanical losses we disregard).

$$p_2=p_1\frac{q_1}{q_2}.$$

When using machine 2 as hydraulic motor, feed Q_1 machine 1 (pump) is less than the expenditure Q_2 hydraulic motor

$$i = \frac{Q_1}{Q_2} \text{ times,}$$

a pressure p_1 machine 1 (pump) under the condition of the equality of the power of machines more pressure p_2 machine 2 (hydraulic motor)

$$\frac{p_1}{p_2} = \frac{Q_2}{Q_1} \text{ times.}$$

HYDRAULIC ACCUMULATORS.

Hydraulic accumulator - the equipment/device, which serves for the accumulation of working fluid, which is located under overpressure, that obtains and giving up working fluid only alternately.

During the application/use of storage battery/accumulators, is represented possible to lower because of the accumulation of hydraulic power in dwelling periods in consumption with its performing aggregate/units of hydraulic systems the lifting power up to average power of the users of hydraulic power or to ensure in systems with the occasional action of users interruptions (pauses) on the work of pump under load.

Since the energy, accumulated in storage battery/accumulator, can be returned (storage battery/accumulator can be discharged) into short time, storage battery/accumulator can short-term develop large power. Therefore the application/use of storage battery/accumulators especially is rational in hydraulic systems with the large peaks of fluid flow rate whose values in certain cases considerably exceed the average/mean fluid flow rate. Thus, for instance, the power, developed with hydraulic engines (actuating cylinders), frequently it exceeds during the application/use of a storage battery/accumulator

Fig. 94. Diagram of the application/use of a gas-hydraulic storage battery/accumulator.

Fig. 95. Diagrams of spring hydroaccumulators.

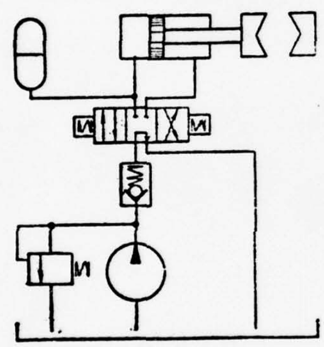

Fig. 94.

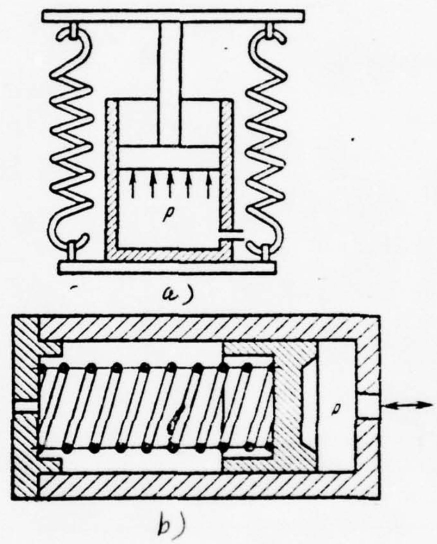

Fig. 95.

the installed power of pump 15-20 times.

Storage battery/accumulator frequently is applied as source of the emergency service of the separate/individual branches of hydraulic system in the failure (or disconnection) of the basic (pump) of power supply. Specifically, such cases include the power supply of the braking system of airplanes and other cargo vehicles. The application/use of storage battery/accumulators has special advantage in the case when is required prolonged period of time any section of hydraulic system to maintain on by pressure (load) in the practical absence in it of fluid flow rate. To is concealed to the cases it is related, for example, prolonged holding under the pressure of the molded parts from rubber and other nonmetallic materials during their vulcanization. In this case, the distributor is establish/installed in the position by which the pump is detached from system and is connected with tank, but the working cavity of actuating cylinder is connected with storage battery/accumulator (Fig. 94).

In machine-building are applied gas (pneumatic) storage battery/accumulators and thinner, predominantly in small pressures, spring.

In spring storage battery/accumulator (Fig. 95) pressure p of liquid is created by the force, which develops with elongation (Fig. 95a) or the compression (Fig. 95b) of springs. The current pressure p will be determined from the expression (friction we disregard)

$$p=\frac{P_{np}}{F},$$

where of the $P_{np}=C(h_0+h)$ - the force of the reduction (elongation) of springs; C is a stiffness coefficient of springs; h_0 and h is a preliminary reduction of springs and their reduction in the process of charge and discharge of storage battery/accumulator.

Since the force of spring depends on its strain, the pressure of liquid in this storage battery/accumulator depends on the degree of its discharge (from amount of liquid in the cylinder of storage battery/accumulator).

Fig. 96. Hydraulic piston type storage battery/accumulator.

Fig. 97. Diaphragm type hydropneumatic storage battery/accumulators.

Key: (1). Gas. (2). Gas cavity. (3). Liquid. (4). Rubber blanket. (5). Liquid.

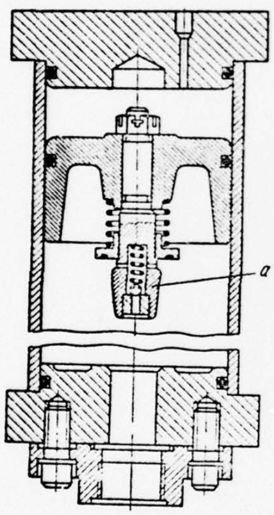

Fig. 96.

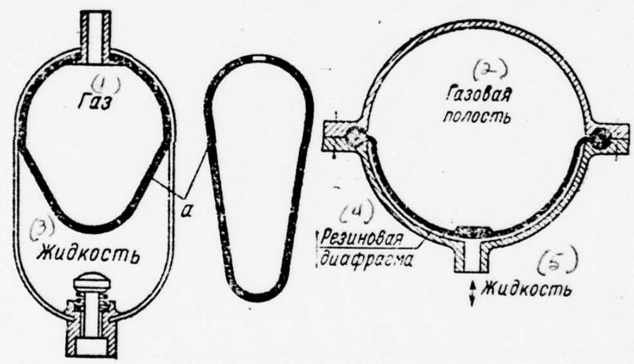

Fig. 97.

In high-pressure hydraulics (200-300 kgf/cm²) are applied the gas-hydraulic storage battery/accumulators, in which the accumulation and the return of energy occurs because of compression and expansion of gas (air or nitrogen).

Gas-hydraulic storage battery/accumulator is the enclosed container, filled by compressed gas with certain initial pressure of the $p_{н}$ of charge.

During the supplying into this container of liquid, the volume gas the chamber decreases, in consequence of which the pressure of gas is raised, reaching toward the end of the charge (filling) by the liquid of p_{max}. The amount of subject into the storage battery/accumulator liquid and the mean pressure of gas for an isothermal process

$$p_{cp} \approx \frac{1}{2}(p_{H} + p_{max})$$

determines the supply of energy (energycontent), which can be in full or in part used with the discharge of storage battery/accumulator.

In the storage battery/accumulators, used in the hydraulic systems of machines, the liquid and gas are usually divided by the piston or other means. The separation of liquid and gaseous media removes the possibility of the dissolution of gas in liquid.

In accordance with the type of the used separator of media, they distinguish piston (Fig. 96) and diaphragm (Fig. 97a and b) storage battery/accumulators. A deficiency/lack in the first is friction of

piston in the cylinder, as a result of which is created hysteresis in the work of storage battery/accumulator. Losses of pressure on the overcoming of the frictional forces of piston reach at the nominal pressure 320 kgf/cm^2 usually 1.5-3 kgf/cm^2. A deficiency/lack in the piston storage battery/accumulators is also the possibility of disturbance/breakdown, in particular under conditions low-temperature, airtightness on the landing place of piston in cylinder.

In order to remove the possibility of the losses of gas in the discharged piston storage battery/accumulator and the inoperative hydraulic system, is applied valve a of automatic opening, which with the arrival of piston at the end position, which corresponds to the discharged storage battery/accumulator, overlaps exit (expenditure) opening/aperture, cutting off in the cylinder of storage battery/accumulator certain amount of liquid.

These deficiency/lacks to a considerable degree are removed in the storage battery/accumulators in which the separation of media is realized with the aid of elastic rubber blanket. They are of gas cylinder (Fig. 97a) and spherical (Fig. 97b) types.

Since in storage battery/accumulator with diaphragm the pressure of gas is transferred virtually directly to the surface of liquid, the latter will be found under that pressure, as gas. Furthermore, since the resistance to deformation of diaphragms is insignificant, these storage battery/accumulators are virtually inertia-free.

For the preservation of diaphragm from extrusion into the opening/aperture of exit branch with the complete discharge of storage battery/accumulator, it is supplied with thickening (Fig. 97b). In the diagram, presented in Fig. 97a for this purpose is applied the valve, which under the action of diaphragm with the complete discharge of storage battery/accumulator by liquid overlaps delivery outlet how is prevented the damage of diaphragm.

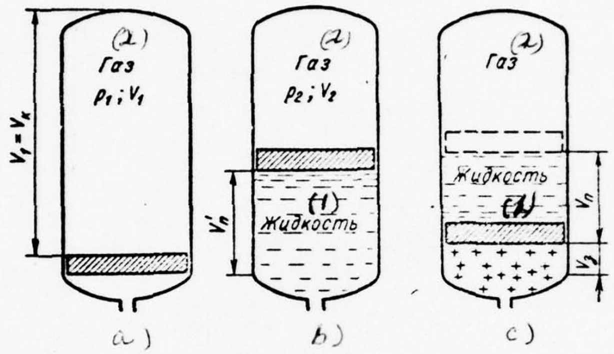

Fig. 98. The design diagram of gas-hydraulic storage battery/accumulator.

Key: (1) Liquid; (2) Gas

Spherical (spherical) type storage battery/accumulators differ from the cylindrical and the gas cylinder in terms of relative compactness and a small mass. The latter is caused by the special feature/peculiarities of spherical form (surface of the container of spherical form is less with the same volume, than the surface of the container of another form), and also fact that in the walls of the container of spherical form, which is located under the pressure of liquid, are created the voltage/stresses 2 times lower than in the riding-crops of cylinder of the same diameter.

Capacity of storage battery/accumulators.

In the calculations of gas-hydraulic storage battery/accumulator by the basic questions are the determination of its structural/design (complete) capacity of V_K and working volume the $V_ж$ of liquid hearth by which is understood the volume of liquid, displaced by gas from storage battery/accumulator in the process of its complete discharge with decompression of gas in the assigned range (interval). The product of working volume and the mean pressure of gas in this pressure range defines the external work (energy) of storage battery/accumulator.

From Fig. 98 follows for an isothermal process

$$\frac{V_2}{V_1}=\frac{p_1}{p_2};\ V_2=V_1\frac{p_1}{p_2};\ p_2=p_1\frac{V_1}{V_2}, \qquad (48)$$

where p_1 and V_1 - initial pressure and the volume of gas to the filling (charge) of storage battery/accumulator with liquid (Fig. 98a); p_2 and V_2 are the stagnation pressure and the volume of gas at the end of the filling (charge) of storage battery/accumulator by liquid (Fig. 98b).

Volume V_1 gas before the filling of storage battery/accumulator with liquid is equal to the structural/design capacity of the V_K of the storage battery/accumulator of $(V_1=V_K)$ (Fig. 98a), and the working volume of the liquid of V_n is equal to a change in the volume of gas with the charge (or discharge) of the storage battery/accumulator:

$$V_n'=V_1-V_2.$$

After substituting into the last/latter expression V_2, we will obtain

$$V_n' = V_1\left(1 - \frac{p_1}{p_2}\right) = V_\kappa\left(1 - \frac{p_1}{p_2}\right).$$

The last/latter expression is correct under the condition of the complete displacement of liquid from storage battery/accumulator with its discharge.

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In practice pressure p_1 is accepted to call the initial (preliminary) pressure of the charge of accumulator by gas (without liquid) and to designate $p_н$, pressure p_2 - by the maximum operating pressure at the end of the charge by its liquid and to designate p_{max}.

In accordance with this, the last/latter expression will take form

$$V_п' = V_к\left(1 - \frac{p_н}{p_{max}}\right).$$

Virtually the discharge of storage battery/accumulator do not lead to complete displacement liquids, but retain in it certain supply the $V_з$ of liquid (Fig. 98c), necessary for providing a

reliable work of the automation of the start of pump for the recharging of storage battery/accumulator, after the pressure as a result of fluid flow rate is lowered to the minimum working value of p_{min}. The structural/design volume of storage battery/accumulator in this case is utilized not completely; storage battery/accumulator will be partially filled by the unproduced volume of the $V_{з}$ of liquid, which lowers its useful capacity. This unproduced volume (supply) must be, if are not produced other requirements, minimum. In accordance with this must be observed the condition of $p_{min} > p_{н}$, whereupon in all cases when selecting the initial air pressure it is necessary to approach its closest approach for the minimum operating pressure of p_{min}.

The process of the compression of gas from the initial $p_{н}$ to the minimum working p_{min} of pressure flow/lasts over those laws, as in the examined case:

$$V_{з} = V_{к}\left(1 - \frac{p_{н}}{p_{min}}\right). \qquad (49')$$

Taking into account the indicated supply (volume) the working volume decreases under otherwise equal conditions by the volume of the supply of V_s and it will be

$$V_n = V'_n - V_s.$$

after substituting into this expression of V_s from expression (49), let us find the working volume of storage battery/accumulator when the $p_{min} > p_H$ and polytropic exponent n = 1:

$$V_n = V_n' - V_s = V_\kappa\left(1 - \frac{p_\kappa}{p_{max}}\right) - V_\kappa\left(1 - \frac{p_\kappa}{p_{max}}\right) = V_\kappa\left(\frac{p_\kappa}{p_{min}} - \frac{p_\kappa}{p_{max}}\right) \quad (50)$$

or

$$\frac{V_n}{V_\kappa} = \frac{p_\kappa}{p_{min}} - \frac{p_\kappa}{p_{max}}. \quad (51)$$

Expressions (50) and (51) show that the useful capacity of the aaaa of storage battery/accumulator depends under otherwise equal conditions on the relation of p_κ/p_{max} and for this p_{max} on the initial pressure of the p_κ of battery charging by gas.

The range of pressure change usually is selected within limits

$$\frac{p_{max} - p_{min}}{p_{max}} \leqslant 0,15.$$

Polytropic process.

With battery charging (filling with liquid) occurs the compression of gas, increase in its temperature and the heat emission through the wall in environment, while with discharge it occurs the expansion of gas, also, as a result of the release in this case of the energy of compressed gas lowering in its temperature and the inflow of heat from without. These processes usually occur at velocities, which correspond to a polytropic change in the state of the gas. For this change in the state (n > 1) expressions (48) take the form

$$\frac{v_2}{v_1} = \left(\frac{p_1}{p_2}\right)^{1/n}; \quad v_2 = v_1\left(\frac{p_1}{p_2}\right)^{1/n} = v_\kappa\left(\frac{p_\kappa}{p_{max}}\right)^{1/n};$$
$$p_2 = p_{max} = p_1\left(\frac{v_1}{v_2}\right)^n = p_n\left(\frac{v_\kappa}{v_2}\right)^n.$$

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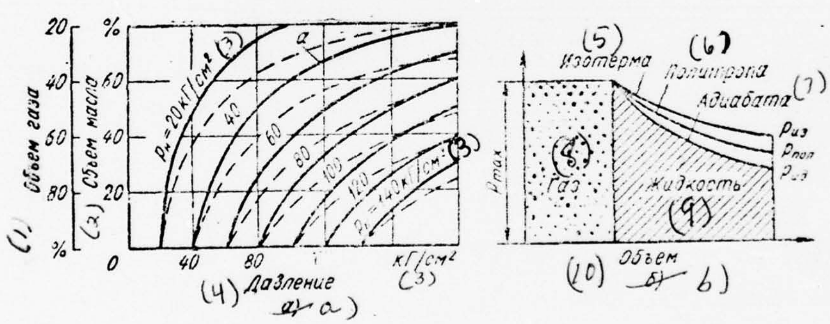

Fig. 99. Curves, that characterize the mode/conditions of the compression of gas in storage battery/accumulator.

Key: (1). Volume of gas. (2). Volume of oil. (3). kgf/cm². (4). Pressure. (5). Isotherm. (6). Polytropic curve. (7). Adiabatic curve. (8). Gas. (9). Liquid. (10). Volume.

In accordance with this

$$\frac{v_n}{v_\kappa} = \frac{v_1 - v_2}{v_\kappa} = \left(\frac{p_\mathrm{н}}{p_\mathrm{min}}\right)^{\frac{1}{n}} - \left(\frac{p_\mathrm{н}}{p_\mathrm{max}}\right)^{\frac{1}{n}}$$

or

$$\frac{v_n}{v_\kappa} = \left(\frac{p_n}{p_\mathrm{max}}\right)^{1/n}\left[\left(\frac{p_\mathrm{max}}{p_\mathrm{min}}\right)^{1/n} - 1\right], \qquad (52)$$

where a is the polytropic exponent whose value is determined experimentally.

During the rapidly elapsing processes the heat exchange with medium virtually is absent and a change in the state of the gas occurs according to adiabatic law with adiabatic index k = 1.4. In accordance with this

$$\frac{V_n}{V_\kappa} = \left(\frac{p_\mathrm{н}}{p_\mathrm{min}}\right)^{\frac{1}{k}} - \left(\frac{p_\mathrm{н}}{p_\mathrm{max}}\right)^{\frac{1}{k}}.$$

Experiment shows that for the standard storage battery/accumulators, designed for pressure 200 kgf/cm², the wall thickness such that the purely isothermal cycle occurs only for the duration of the process of charge or discharge not less than 3 min. For the duration of cycle, equal or it is less than 0.5 min, more preferable to apply adiabatic process (n = 1.4). Taking this into account in practice the polytropic exponent for the widespread storage battery/accumulators and the mode/conditions of their work (time is equal approximately 0.5 min) is accepted in average/mean n = 1.3.

Effect of the mode/conditions of compression and expansion of gas.

The temperature changes, which proceed during compression and expansion of gas in mode/conditions $n > 1$, can lower the useful capacity of storage battery/accumulator. The latter is visually evident from expression (52), which shows that the volume of liquid in storage battery/accumulator with $n = 1$ it will be more than with $n > 1$.

As the illustration of the latter Fig. 99a gives pressure curves in function of the compression of gas for $n = 1$ (solid lines) and $n = 1.4$ (the broken lines), which show that for a pressure increase from initial value of 40 to 100 kgf/cm² in isothermal process ($n = 1$) the volume of gas chamber must be decreased by 60o/o of initial value (see the curve of a), whereas during adiabatic process ($n = 1.4$) this pressure will be reached during a decrease in the volume of gas only by 48o/o of its initial value. Consequently, the volume of liquid in storage battery/accumulator with $n = 1$ it will be more than with $n = 1.4$, so, as it will be more, also, at any value of $n > 1$.

Figure 99b gives the performance record of storage battery/accumulator during the equal mode/conditions of discharge

(release of energy). The points of $p_{\text{из}}$, $p_{\text{пол}}$ and $p_{\text{ад}}$ express the stagnation pressure of gas with discharge from p_{max} on isotherm, polytropic curve and adiabatic curve.

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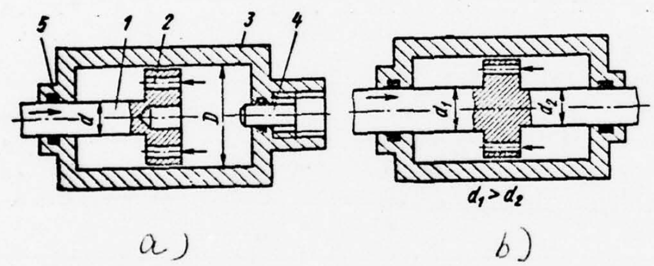

Fig. 100. Diagrams of liquid springs.

The shaded area/site expresses work during the adiabatic mode/conditions of discharge.

LIQUID SPRINGS.

The true liquid is the elastic body, which are subordinated with certain approach/approximation to the law of the compression of Hooke (change in the volume of liquid during its compression it is proportional to pressure change), which makes it possible to utilize liquids for the construction of powerful springs and shock absorbers.

The schematic diagram of liquid spring is given in Fig. 100a. Spring consists of cylinder 3 and entering it through the sealing node/unit stock/rod 1 with piston 2, employed direction for the latter. Cylinder 3 is filled with the deaerated liquid under certain initial pressure p_1, which determines the force of the initial "compression" of spring, computed (not allowing for friction) according to expression

$$P=p_1f,$$

where $f = \pi d^2/4$ - the sectional area of stock/rod 1 with a diameter of d.

With stock/rod's countersinking 1 into cylinder 3, pressure of liquid as a result of its compression will be raised, after achieving toward the end of the course of stock/rod p_2, determined the compression ratio of liquid (by change in the cylinder capacity 3), and also by the coefficient of the compressibility of the latter.

For providing a hardness of construction and possibility of obtaining simultaneously with this large courses are applied the diagrams, based on differential stock/rod (Fig. 100b). As stock/rod's working (not balanced) area here serves a difference in the areas of his left and right sections:

$$f=\frac{\pi}{4}(d_1^2-d_2^2).$$

The characteristics of liquid spring in essence depend on the coefficient of compressibility β (or from bulk rigidity modulus) liquid.

On the basis of expression (5)′ (change) in the pressure differential at the end and the beginning of compression

$$\Delta p = p_2 - p_1 = \frac{1}{\beta} \cdot \frac{\Delta V}{V_1}; \quad p_2 = p_1 + \Delta p = p_1 + \frac{1}{\beta} \cdot \frac{\Delta V}{V_1}, \qquad (53)$$

where $\Delta V/V_1$ is a relative change in the volume of liquid during a change in the pressure Δp; p_1 and p_2 is the initial pressure is of liquid (before compression of spring) andpressure at the end of the compression; β is the average for given pressure range coefficient of the compressibility of liquid.

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Thus, a change in the volume of liquid with the change of

pressure on Δp

$$\Delta V = V_1 - V_2 = V_1 \Delta p \beta,$$

where V_1 and V_2 are a volume of liquid at the initial pressure (is equal to cylinder capacity 3) and its volume during a change in the pressure on Δp.

In accordance with this force on the stock/rod of liquid spring at the end of its compression in isothermal process (not allowing for friction)

$$P_2 = fp_2 = f\left(p_1 + \frac{1}{\beta} \cdot \frac{\Delta V}{V_1}\right).$$

From this equation it follows that since the coefficient β compressibility depends on pressure, decreasing with an increase in the latter, liquid spring possesses alternating/variable hardness in the course of compression - with compression spring constant

increases.

For the spring whose diagram is represented in Fig. 100a, a decrease in the volume of liquid during compression can be expressed (not allowing for the deformation of container)

$$\Delta V = fh, \tag{54}$$

where f and h are a sectional area and piston stroke (plunger).

In accordance with this, expression (53) during isothermal compression of spring to size/dimension of h with the initial pressure of charge p_1 will take form

$$p_2 = p_1 + \frac{1}{\beta} \cdot \frac{fh}{V_1},$$

a stock/rod's course for this change in the pressure

$$h = \beta (p_2 - p_1) \frac{V_1}{f}.$$

The advantages of liquid springs include also the simplicity of the provision for the assigned force of the pretightening of spring. The latter usually is realized by screw/propeller 4 (Fig. 100a) whose shank with screwing up compressible liquid up to a pressure of p_1. The section of the f_x of shank and its displacement/movement of aaaa for a pressure increase from zero to the assigned initial pressure p_1 compression of spring can be determined from relationship (54):

$$f_x h_x = \Delta V = p_1 V_1 \beta; \quad f_x = \frac{p_1 V_1 \beta}{h_x};$$

$$p_1 = \frac{f_x h_x}{V_1 \beta}.$$

Compression work of liquid.

By the important parameter, which characterizes the state of liquid, which is located under high-pressure action, is the work of its compression, which determines the potential energy of the compressed liquid. With the previously accepted assumption that the liquid obeys Hooke's law itself and $p_1 = 0$, the energy possibilities of volume of liquid V_1, compressed up to a pressure of p_2 in rigid container (not allowing for vasodilation), can be expressed in isothermal process by equation

$$A = p_{cp}\Delta V = p_{cp}hf, \qquad (55)$$

where A - the energy (work) of the compressed liquid; p_{cp} - average in the process of the compression of the pressure of liquid

$$(p_2 > p_{cp} > p_1).$$

Under the assumption that the compression of liquid obeys Hooke's law itself, mean pressure can be approximately accepted

$$p_{cp} = \frac{p_2 + p_1}{2}.$$

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In accordance with this, expression (55) will take form

$$A = \frac{p_2 + p_1}{2} \Delta V.$$

On the basis of expression (53)

$$\Delta V = V_1 (p_2 - p_1) \beta,$$

taking into account what we will obtain

$$A = \frac{1}{2} (p_2^2 - p_1^2) V_1 \beta.$$

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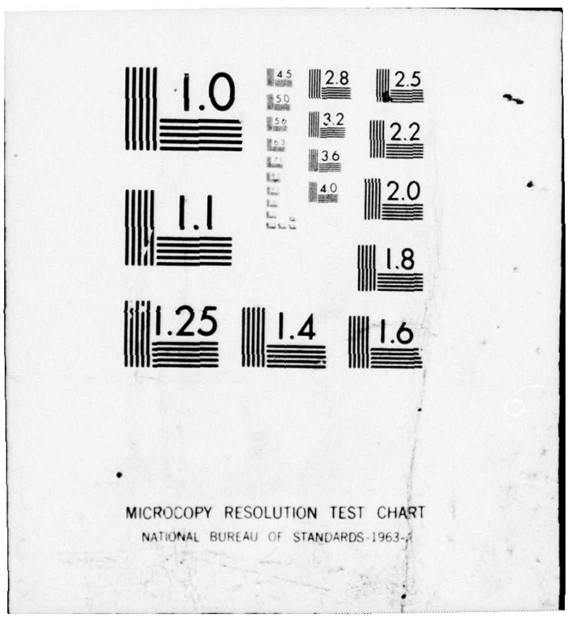
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For the zero pretightening of spring ($p_1 = 0$) we will have

$$p_{cp} = \frac{p_2}{2}; \quad \Delta p = p_2$$

and volume change [see expression (53)]

$$\Delta V = V_1 p_2 \beta.$$

In accordance with this, we will obtain for condition $p_1 = 0$ approximation for the calculation of the energy of the compressed liquid

$$A_{max} = \frac{p_2}{2} \Delta V = \frac{1}{2} p_2^2 V_1 \beta. \tag{56}$$

The taken in the last/latter expressions condition

$$p_{cp} = \frac{p_2}{2}$$

approximately valid only for the relatively small (to 600-800 kgf/cm²) pressures, within limits of which the compression and the pressure of liquid are connected by dependence, virtually close to linear (see Fig. 9a).

From expression (56) it follows that for obtaining maximum work at the assigned pressure p_2 the compression of liquid it is necessary to have under otherwise equal conditions its largest possible initial volume V_1 or with its assigned initial volume the maximum pressure p_2. In equal measure from this point of view it is expedient to select the liquids, which possess the maximum coefficient of compressibility β (with the minimum elastic modulus) and its minimum dependence on different factors and, in particular, from pressure and temperature, and also by the low temperature coefficient of expansion.

From the existing liquids these conditions most completely satisfy their silicone ethylpolysiloxane) brands whose compressibility approximately by 40-50o/o is higher than the compressibility of the liquids of mineral origin (see Fig. 9b).

to account for deformation under the pressure of the liquid of the walls of cylinder the calculations must be conducted with the replacement of the bulk rigidity modulus of liquid E by the given

bulk modulus of the liquid of E_{np} in the elastic conduit/manifold:

$$E_{np} = \frac{E_{\text{ж}}}{1 + E_{\text{ж}} \frac{D}{E_{mp} s}},$$

where D and s are a diameter and the wall thickness of conduit/manifold; E_{mp} - the modulus of elasticity of the material of the walls of conduit/manifold (cylinder).

Effect on the characteristic of the spring of the mode/conditions of its compression and quality of liquid.

Above are examined the questions of the compression of liquid spring concerning the isothermal process which is characterized by so slow a change in the volume, that the isolatable in this case heat is scattered, as a result of which compression occurs at constant temperature of liquid.

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However, the compression of liquid in here the case of its application/use in question usually flow/lasts with the speeds at which the isolating heat completely is not scattered, partially it is concentrated in liquid, raising its temperature and increasing volume, or changing its other characteristics. In view of this the pressure of liquid during compression of spring at real speeds can exceed pressure during its compression under the isothermal conditions.

Taking into account this during refined calculations of the high-speed liquid springs, one should proceed not from isothermal, but from the polytropic process during which the developing during the compression of liquid heat partially is expend/consumed on an increase in its temperature. Maximum from this viewpoint is the process of the compression during which entire heat, which corresponds to the energy of the compression of liquid, is expend/consumed on an increase in its temperature. The calculations show that during compression on this maximum process of mineral liquid from zero pressure to 3500 kgf/cm² an increase in the temperature is approximately equal to 35°C.

Since an increase in the temperature of liquid is accompanied by fall in the bulk modulus of its elasticity, and also by the temperature expansion of liquid, the characteristic of spring at the end of its compression in this mode/conditions can differ from the calculated in isothermal mode/conditions. However, since the increase in the temperature of liquid, which proceeds during the ram compression of spring, is accompanied by a fall in the modulus of elasticity of liquid and simultaneously by an increase in its volume, and consequently, in the initial pressure increase of the charge of spring, temperature effect on one of these parameters partially compensated by opposite influence on another parameter, as a result the difference in characteristics during static and ram compression usually small.

Attenuation of energy.

In the liquid springs, utilized as shock absorbers, is provided for the attenuation of certain part of the energy of the compression of liquid, for which spring they supply with damper in the form of

throttle/choke one- or bilateral action. The attenuation of energy (throttling/choking of liquid) occurs either about foreward stroke (during compression of spring), or with back stroke (with the straightening of spring), or simultaneously both with straight line and with back strokes.

Distributed are second type liquid springs, in which the attenuation of energy (braking) occurs with the straightening of spring. This is realized in that which is examine/considered by us to diagram fact that the liquid, included in the chamber, is pressed with the straightening of the spring through eyelets in floating overlapping valve 1 (Fig. 101a). This valve during compression of spring is moved by the pressure of the displaced liquid to the left and open/discloses passage openings a in piston 2, thanks to which liquid it flow/lasts in this case without resistance. With the straightening of spring (Fig. 101b) valve under the action of the flow of compressible liquid is moved and overlaps to the right passage openings, as a result of which the displacement of liquid from the left cavity of cylinder into right occurs only through choke opening/apertures in the valve of 1 small section, with course through which certain part of the energy of the compressed liquid is converted into heat.

In view of the complexity of the process of attenuation the calculation produces, accepting a series of assumptions and averaging the value of the unknown parameters.

After subdividing work (energy) A, which must to absorb the liquid with the passage through the choke channels (opening/apertures) of damper, on the stroke of spring h, let us find the average/mean force P on its stock/rod developed with a jump/drop in the pressure, created by resistance of these channels:

$$P = \frac{A}{h}.$$

Fig. 101. Diagrams of the liquid springs: a and b) by damper; c) pulse hydraulic drive.

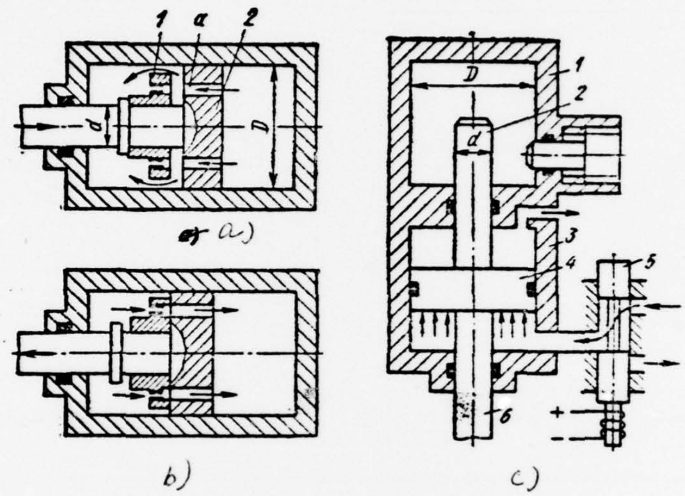

Fig. 101.

Inasmuch as

$$P = f\Delta p_{\partial p}$$

where f is an area of the stock/rod of spring; $\Delta p_{\partial p}$ - the resistance of throttle/choke,

we can write

$$\Delta p_{\partial p} = \frac{A}{hf}.$$

Jump/drop in the pressure of $\Delta p_{\partial p}$ on throttle/choke and fluid flow rate through the choke opening/aperture of section ω are connected by relationship [see expression (20)]

$$Q = \mu\omega\sqrt{\frac{\Delta p_{\partial p}}{\rho}2},$$

where Q - the average fluid flow rate; μ is a coefficient of expenditure/consumption; ρ - the density of liquid.

After assigning time T of the straightening of spring to length h, we find the average speed of the v_n of the motion of its piston:

$$v_n = \frac{h}{T}.$$

In accordance with this the average expenditure/consumption Q of the liquid through the choke opening/aperture of damper (valve)

$$Q = v_n \frac{\pi}{4} (D^2 - d^2),$$

where D and d are a diameter of cylinder and stock/rol of shock absorber.

Solving the together given equations, we find the sectional area of choke opening (we allow/assume the complete airtightness of other points of connection)

$$\omega=\frac{\dot{Q}}{\mu\sqrt{\frac{2\Delta p_{\partial p}}{\rho}}}=\frac{\pi v_n(D^2-d^2)}{4\mu\sqrt{\frac{2\Delta p_{\partial p}}{\rho}}}.$$

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The coefficient of expenditure/consumption for a diaphragm throttle/choke (in the form of opening/aperture in fine/thin wall under condition $s \leqslant 2d$, where s and d are wall thickness and the diameter of opening/aperture), can be accepted $\mu = 0.62$.

Average pressure differential during the short-cut calculations

$$\Delta p_{\partial r} = \frac{p_{max}}{2}.$$

In diagrams with attenuation on foreward stroke, the float valve is arrange/located from opposite side of piston.

Pulse hydraulic drive.

The elastic properties of liquid are utilized for the creation of pulse hydraulic drive which is applied in the machines of percussion (hammers and other installations), and also as the source of vibrations in testing units. Similar drive makes it possible to obtain to 300-400 momentum/impulse/pulses (courses) per minute, with small courses the number of momentum/impulse/pulses (vibration frequency) can reach 100 per second.

The operating principle of this drive (Fig. 101c) is based on the use of an energy of the instantaneous expansion of the precompressed liquid, whereupon with the release of this energy in short time (0.005-0.01 s) it is possible to obtain large power with relatively small pressures and the volumes of compressible liquid [see expression (56)].

Drive consists of the liquid spring, which is those filled by liquid under certain initial pressure p_1 container (cylinder) 1 together with its attached plunger (rod) 2, connected with the piston 4 of a power cylinder 3. Piston 4 will bear from the side, opposite to plunger 2, stock/rod 6, to whom is connected external load. The power supply of actuating cylinder 3 by liquid is realized with the aid of distributor 5 with the high-speed (electromagnetic or others) drive with the aid of which the working (lower) cavity of cylinder consecutively is connected with power supply (pump) and with tank. During the supplying of liquid into actuating cylinder, plunger 2 steps down, compressible liquid in container 1 (compression pressure usually 600-1000 kgf/cm². During the changeover of distributor 5 at the position of bleeding from the working cavity of cylinder 3, plunger 2 with connected load fast and well is moved under the effect

of pressure of the compressed liquid down.

Distributor 5 is designed for the release of the energy of the liquid, compressed in container 1, in the shortest time (0.005-0.006 s), thanks to which it is represented possible to obtain with the small dimensions of drive large instantaneous power.

VELOCITY CONTROL OF HYDRAULIC ENGINES.

Depending on the method by which is achieved a change in the supply of the liquid, directed in hydraulic engine, they distinguish of two basic method of the control of its speed: choke and volumetric.

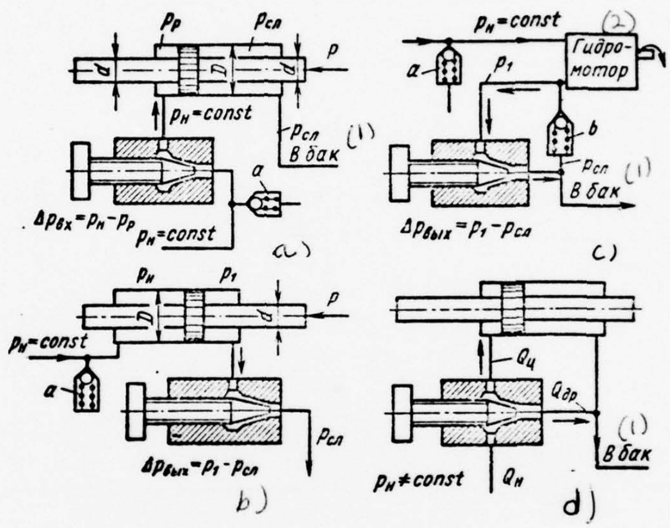
With throttle control a change in the feed, directed in hydraulic engine, is realized by a change in the hydraulic friction of the line in which it is included, and by the diversion/tap (gutter) of part of the feed of pump in tank.

With volumetric control a change in the supply of the liquid, which enters the hydraulic engine, is achieved by a change in the working volume or frequency of the rotation/revolution of pump, i.e., volumetric control predetermines application/use in the hydraulic system of the adjustable pump or pump drive with the adjustable frequency of rotation/revolution.

Are common both first and the second methods, whereupon the first of these methods is applied predominantly in the hydraulic systems of power-handling capacities (to 5-10 h.p.) and by the second - at higher powers.

Fig. 102. Diagrams of the throttle control of the speed of hydraulic engine.

Key: (1). In tank. (2). Hydraulic motor.



Throttle velocity control of hydraulic engines.

The throttle/choke is the simplest and most widely accepted speed governor of hydraulic engine. The major advantage of hydraulic drive with throttle control is the possibility of a smooth change in the speeds, the simplicity of control of distributor, and also the fact that the forces, required for a control, they can be lowered during application/use two- and of three-stage intensification to 2-3 g. These advantages are especially valuable in the systems of automation, since they allow/assume the application/use of low-power signals.

Throttle control is applied predominantly in diagrams with the hydraulic generator of the constant pressure of $p_н = \text{const}$.. As this generator can serve any uncontrolled pump, equipped with overflow (restricting pressure) valve, or the adjustable pump with the equipment/device, which restricts pressure by changing the expenditure/consumption. This same requirements for $p_н = \text{const}$ partially satisfies the hydroaccumulator which supplies system in the periods of the disconnection of pumps.

Depending on the form of power supply the existing systems of throttle control can be divided into the systems, supply of power (pump) which it has:

alternating/variable expenditure/consumption ($Q \neq \text{const}$) and the constant pressure of the ($p_н \neq \text{const}$) of liquid, determined by the control of overflow valve a (Fig. 102a and b);

constant flow rate/consumption ($Q = \text{const}$) and the variable pressure of ($p_н = \text{const}$), determined by the load of hydraulic engine (see Fig. 102d).

More rarely are applied combined system.

Is most common the first system, which has special advantages when for several hydraulic drive is applied one supply of power (pump).

Installation diagrams of throttle/choke. Throttle/choke

(regulator) in systems from $p_H = \text{const}$. is establish/installed either in the feed line of hydraulic engine (at inlet), as shown in Fig. 102a and 103a or in drain line (at output/yield), as shown in Fig. 102b and 103b. Both in that and in another case it is represented possible to obtain the speeds of hydraulic engine from the zero to the maximum. The excess of the liquid, applied by pump, is abstract/removed into the tank through the overflow valve.

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Estimating these mounting methods of throttle/choke, it should be noted that diagram with throttle/choke in drain line (see Fig. 102b) they provide the bilateral hardness of the supplied hydraulic engine. Therefore these systems can be used with oscillating and regular external loads of the latter for which diagram with the regulator, established/installed on feed line (see Fig. 102a), are less suitable, since during a sign change of the load of hydraulic engine the speed of the motion of its exit stock/rod (or shaft) it can be raised, since this throttle/choke here it does not resist.

In equal measure the system with throttle/choke at inlet is

unworkable in the mode of the high accelerations of exit stock/rod or shaft. From the diagram, presented Fig. 102a it is apparent that shows during a sharp reduction in the feed of liquid into cylinder by means of throttling/choking at the inlet into it the piston is moved under the action of the force of inertia of the driving mass, creating in this case in working cavity evacuation/rarefaction, i.e., occurs the discontinuity of the continuity of the flow:

$$v_{nop} \neq \frac{Q}{f}.$$

The latter has especially important value for diagrams with the hydraulic engine of rotary motion (with the hydraulic motor), which can work in transient conditions with high speed and the accelerations of output shaft, in this case the force of inertia of the rotating parts of the hydraulic engine with the connected to it mass of external load they can reach the significant magnitude. During the unit of throttle/choke in drain line an increase in the velocity of output shaft impedes the friction of this throttle/choke,

which will be raised proportional to the square of velocity. However, during the abrupt decelerations of hydraulic motor in this diagram in line between it and the throttle/choke, can arise high pressures. For the preservation of system and hydraulic motor from an inadmissible pressure increase in this line, is establish/installed the safety valve b (Fig. 102c).

Furthermore, the diagrams with throttle governor in drain line are more stable against auto-oscillations than diagram with regulator in feed line, and in particular at the low speeds of the motion of hydraulic engine, which is caused in essence by the more intense attenuation (scattering) of vibrational energy.

Furthermore, the unit of throttle/choke at output/yield is more preferable its unit at inlet due to the probability of occurence in the system of an abrupt change in the load of the actuating cylinder, possible, for example, on leaving of the given with the aid of this cylinder instrument from article. In this case the energy of the compressed working fluid and elastically deformed mechanical cell/elements is free/released, which can cause the hydraulic impacts which with the throttling/choking of liquid on gutter considerably are smoothed.

The equilibrium condition of piston in the diagram of actuating cylinder with the bilateral stock/rod of equal sections during the unit of throttle/choke at inlet (see Fig. 102a)

$$p_p F = p_{c\Lambda} F + P + T.$$

Since $p_p = p_н - \Delta p_{вх}$, we can write

$$\Delta p_{вх} = p_н - p_{c\Lambda} - \frac{P \pm T}{F}, \qquad (57)$$

where the $p_p < p_н$ - operating pressure in cylinder (see Fig. 102a), the determined by load $p_н = \text{const}$ - applied pressure (pump), determined by the adjustment of overflow valve; $\Delta p_{вх}$ - the pressure differential on input throttle/choke; $p_{c\Lambda}$ - pressure in drain line; P is the load, applied to the stock/rod of hydraulic engine (actuating cylinder); T - frictional force in cylinder; $F = \frac{\pi(D^2 - d^2)}{4}$ - the effective area of cylinder; here D and d are a diameter of piston and stock/rod.

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During the unit of throttle/choke at output/yield (see Fig. 102b) indicated dependence take the form

$$p_{н}F = p_1F + P \pm T.$$

Since $p_1 = \Delta p_{вых} + p_{сл}$, it is possible to write

$$\Delta p_{вых} = p_{н} - p_{сл} - \frac{P \pm T}{F}, \qquad (58)$$

where p_1 is a counterpressure in the nonoperative cavity of cylinder; $\Delta p_{вых}$ - jump/drop on output choke.

From expressions (57) and (58) it follows that jump/drop in the pressure on throttle/chokes and, consequently, also the fluid flow rate through them it will change in both diagrams with a change in load P on the stock/rod of hydraulic cylinder, whereupon these the pressure differentials will be under otherwise equal conditions equal between themselves both in the diagram with throttle/choke at output/yield and at inlet. In accordance with this, the dependence of the piston speed on load in both these diagrams will be identical.

Since the pumps of similar systems a $p_{н} = const$ consume independent of the load of servomotor the maximum power, which corresponds to the complete feed of pump and to the maximum pressure, determined by the adjustment (control) of overflow valve, both examined systems of throttle control possess relatively low efficiency.

The used in these systems pumps of constant feed are selected from the calculation of maximally possible for this application/use consumption of liquid by hydraulic engine, and consequently, at the low speeds of hydraulic engine the surplus of the feed of pump, expendable under the pressure of the $p_{н}$ through the overflow valve, it composes the considerable part of the feed of pump. In the worse from this viewpoint case when the speed of hydraulic engine is close to zero, virtually entire the applied on pump inlet constant power is expend/consumed on heating liquid with its overflowing through the overflow valve, adjusted to the maximum pressure. This it conducts to the overheating of liquid, which is accompanied by the loss by it of some properties.

Systems with variable pressure. More rarely are applied systems with variable pressure ($p \neq$ const), in which the pressure of power supply is determined by the load of hydraulic engine.

Figure 102d (see also Fig. 103c) gives the diagram of the similar hydraulic system in which the throttle/choke is connected in parallel to hydraulic engine (actuating cylinder). The excess of liquid in this diagram is abstract/removed (is expended) into tank not through the overflow valve, as this occurred in previously examined systems (see Fig. 102a and b), but through this throttle/choke, established/installed in parallel with hydraulic engine (on the line, which connects the main line applied pressure with tank).

The liquid, applied by pump in the volume of the $Q_н$, is divided in this diagram into two parallel flows, one of which $(Q_ц)$ enters actuating cylinder (hydraulic engine), and another $(Q_{др})$ is recasted through the throttle/choke into tank, whereupon these flows inversely proportional to the friction (loads) of the branches:

$$Q_н = Q_ц + Q_{др}.$$

After designating the hydraulic friction of the throttle/choke by

$$r_{\partial p} = \frac{Q_{\partial p}}{\Delta p_{\partial p}},$$

where the $Q_{\partial r}$ and the $\Delta p_{\partial r}$ - the expenditure/consumption and jump/drop of pressure in it, and disregarding pressure in drain line, we can write

$$Q_u = \frac{Q_н F r_{\partial p} - P \pm T}{F r_{\partial p}} = Q_н - \frac{P \pm T}{F r_{\partial p}}.$$

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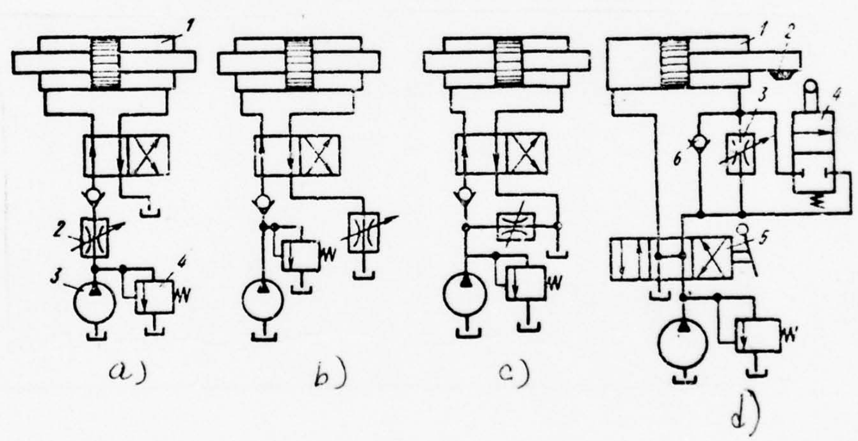

Fig. 103. Circuit diagrams of throttle/chokes into hydraulic system.

Since the power, consumed by pump, depends in this case on the load of performing hydraulic engine (pressure during a dropping of load decreases and during increase increases), the last/latter system arrange/locates less margin of power, which favorably shows up in the stability of its work, and also improves the energy working conditions of system and raises its efficiency.

A deficiency/lack in the last/latter system is the lowered/reduced hardness and the need for individual for each user supply of power (pump). Furthermore, the accuracy/precision of velocity control and its stability with this method of control are lower than in the preceding/previous diagrams from $p_н = \text{const}$, however, even, the heating of liquid less. The latter is caused by the fact that the pressure in this system is proportional to load, and only at its maximum value pressure reaches the value at which is adjusted the overflow valve of pump.

The diagrams of hydraulic systems with the throttle control of the speed of hydraulic engine are given in Fig. 103, whereupon in the diagram, presented in Fig. 103a, throttle/choke is established/installed in delivery line (see also Fig. 102a), in diagram in Fig. 103b - in the line of the gutter (see also Fig. 102b) of willows to diagram in Fig. 103c - between the pressure and drain line (see also Fig. 102d).

By changing the resistance in thefeed circuit of the hydraulic engine with the aid of an adjustable throttle 2 (see Fig. 103a), it becomes possible to limit the flow of liquid to the hydraulic engine (power glider) 1, and, consequently, to regulate the speed of motion of its output unit. Excess liquid supplied by the pump 3 is drawn off (discharged) in this diagram through an overflow valve 4 into a reservoir (tank).

By changing the bypass (by-pass) lines, included with the aid of mechanical (detents on motion work) or electromagnetic equipment/devices, is represented possible to disconnect/turn off throttle/choke on certain part of the way of the piston stroke of cylinder, providing the uncontrolled fast motion of piston. The diagram of a similar hydraulic system with the unit of adjustable throttle/choke 3 in drain line is represented in Fig. 103d. With foreward stroke (to the right) the liquid through distributor 5 is supplied to the left cavity of cylinder 1. From the opposite cavity of cylinder, it is displaced into tank either through adjustable throttle/choke 3 or in the embedded position of two-pass distributor (bypass) 4 into the circuit/bypass of throttle/choke 3 directly into tank.

Fig. 104. Types of throttle governors.

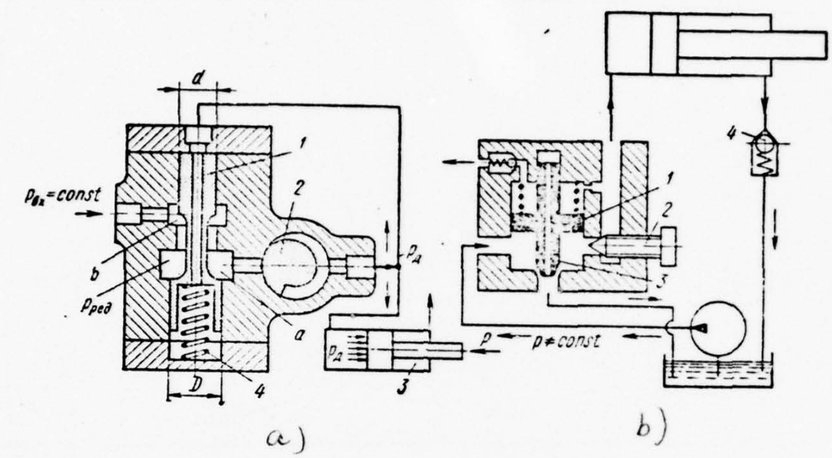

Fig. 104.

Bypass 4 is changed over (it is meltted down) with the aid of detents 2 on the motion work of the cylinder, its embedded position corresponding to the quick traverse of the piston of actuating cylinder 1, which is utilized, for example, in machine tools for the in-rapid traverse of instrument to workpiece and etc.

Piston speed for this case

$$v=\frac{4Q_{H}}{\pi D^{2}}.$$

Piston stroke at this speed continues, thus far detent 2 holds throttle/choke 3 in the embedded position. On the cessation of the action of governing detent, this valve under the force of spring will be establish/installed at the position in which the way of the abstract/removed from cylinder 1 liquid will be overlapped by the adjustable throttle/choke.

During the supplying of the liquid through distributor 5 into the right cavity of actuating cylinder, 1 (it corresponds to the reverse/inverse nonoperative piston stroke of actuating cylinder) liquid enters cylinder into the circuit/bypass of throttle/choke 3 through are reverse/inverse valve 6. The speed of piston stroke in this case

$$v = \frac{Q_H}{F_n} = \frac{4Q_H}{\pi (D^2 - d^2)},$$

where D and d are a diameter of piston and its stock/rod.

Throttle governors with a constant pressure differential. In order to eliminate the effect on fluid flow rate and, consequently, also on the velocity of the hydraulic engine of the load of the latter, are applied the automatically acting throttle governors, which make it possible to ensure during load change virtually constant the pressure differential and constant, other conditions being equal, fluid flow rate through the throttle/choke.

The schematic diagram of this regulator, intended for unit into force main, is represented in Fig. 104a.

Regulator consists of two throttle/chokes 1 and 2, connected of common/general/total housing, from which throttle/choke 2 has constant (manual) resistance tuning (the pressure differential with constant flow rate/consumption Δp_2 = const), throttle/choke 1 is reduction valve with automatic resistance tuning (the pressure differential $\Delta p_1 \neq \text{const}$) depending on the reduced pressure of p_{ped} at output/yield from it and load P of hydraulic engine 3.

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Liquid with the constant the inlet pressure of $p_e = \text{const}$ enters (from pump or another power supply) through input window b and the slot of automatic throttle/choke into annular groove (chamber) a between two bands of the plunger of throttle/choke 1 and from it - toward throttle/choke 2 of fixed resistor, passing through which is

directed to toward hydraulic engine 3.

For the given diagram the outlet pressure of throttle/choke 2 is the operating pressure of the p_∂ of hydraulic engine, which it is equal (friction in cylinder we disregard)

$$p_\partial = \frac{P}{F},$$

where P - the external load, applied to the stock/rod of actuating cylinder; F is the effective area of actuating cylinder.

The plunger of throttle/choke 1 in this diagram is located under the action of the force of spring 4 and of the force of pressure of p_∂ on the unbalanced area

$$\Delta f_n = \frac{\pi (D^2 - d^2)}{4},$$

attempting to displace it toward the position of maximum discovery/opening passage opening b, and by that counteracting by it the forces of the reduced pressure, equal $p_{ped} = \Delta p_2 + p_d$ on the same area where Δp_2 is the pressure differential (friction) in throttle/choke 2.

During steady-state mode

$$p_{ped} \Delta f_n = P_{np} + p_d \Delta f_n$$

or

$$p_{ped} = p_d + \frac{P_{np}}{\Delta f_n},$$

where the P_{np} - the compression of spring 4.

On the basis of the last/latter expressions it is possible to write

$$\Delta p_2 + p_\partial = p_\partial + \frac{P_{np}}{\Delta f_n},$$

whence the pressure differential in throttle/choke 2

$$\Delta p_2 = \frac{P_{np}}{\Delta f_n} = \text{const.}$$

It is obvious, under the condition of the preservation/retention/maintaining of the constancy of a jump/drop in the pressure Δp_2 by constant it will be under otherwise equal

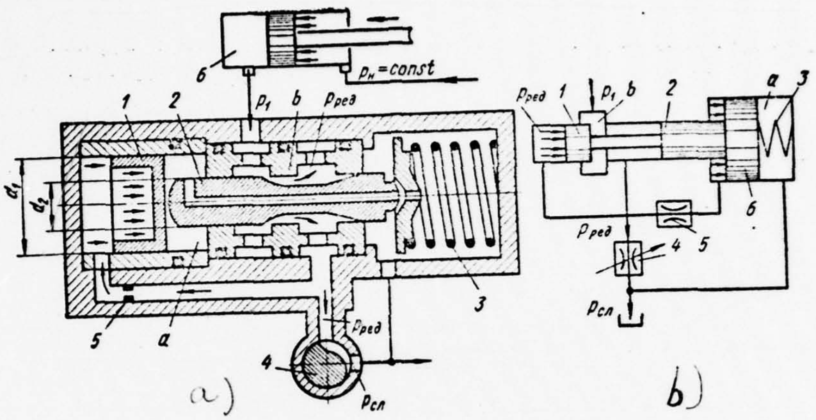
conditions and fluid flow rate independent of load P of hydraulic engine and inlet pressure of the $p_{вх}$, which must exceed $p_{д}$. It is not difficult to see that the increase in the pressure of $p_{д}$, caused by an increase in load P, will unbalance of throttle/choke 1, and it, after moving upward, it will increase passage slot (decreases the pressure differential Δp_1), compensating for the indicated change in the load.

During a decrease in load P of hydraulic engine 3, process flow/lasts in reverse order. The structural and conditional diagram of similar type analogous regulator, intended for unit in drain line, are given in Fig. 105a and b. Regulator is reduction valve (throttle/choke with automatic adjustment) 2 in combination with consecutive throttle/choke 4 of constant adjustment.

The reduced pressure of the $p_{ред} = p_{вх} - \Delta p_2$, where the $p_{вх}$ - pressure at the inlet into reducer and Δp_2 - the pressure differential on reduction piston, acts through piston 1 on inner valve 2, spring-loaded 3, the compression of which determines this pressure.

During change in any reason (load variation, etc.) for the pressure of p_{ax} at the inlet into the regulator, which in this diagram is the counterpressure of the liquid, abstract/removed from the nonoperative cavity of hydraulic engine, it will change instantaneous fluid flow rate through the slotted channel b of automatic throttle/choke, which will cause a change in the friction of throttle/choke 4 and, consequently, also a change in the reduced pressure of $p_{ред}$.

Fig. 105. Construction and the diagram of throttle governor with a constant pressure differential.



As a result of this, the equilibrium of valve 2 will be destroyed, and it will move into the new position in which the losses (jump/drop) of the pressure of Δp_2 in the slotted channel b of throttle/choke again will be equal to a difference in the new input $p_{вх}$ and constant reduced $p_{ред}$ of the pressure of $(\Delta p_2 = p'_{вх} - p_{ред} = \text{const})$.

The reduced pressure of $p_{ред}$ not allowing for the frictional forces, and also action of flow forces of fluid flow to valve 2 and the pressure, caused by the friction of the drainage line of $(p_{сл}=0)$, will be determined from expression

$$p_{ред} = \frac{P_{пр}}{f_п} \approx \text{const},$$

where the $P_{пр}$ - the force of the tightening of spring 3; $f_п$ - piston clearance (for a diagram in Fig. 105b this area is equal the $f_п = f_1 + f_6$, where f_1 are equal the sectional area of piston 1 and f_6 - the effective area of piston 6).

For an $p_{сл} = 0$ the value of $p_{ред}$ is the pressure differential Δp_4 on throttle/choke 4:

$$\Delta p_4 = p_{ред} - p_{сл} = p_{ред} = \text{const.} \tag{59}$$

In accordance with this, the fluid flow rate through throttle/choke 4 will be also constant [see equation (20)].

However, with $p_{сл} > 0$ the pressure differential Δp_4 on throttle/choke 4 decreases by $p_{сл}$, which with respect will have effect on fluid flow rate through throttle/choke 4.

In order to remove effect on the regulator of drainage pressure, drain line they connect with cavity a (in the diagram, shown with Fig. 105a, this is fulfilled with the aid of axial channel in valve 2). In this case will be provided for a constant pressure differential Δp_4 on throttle/choke 4 during all possible changes in the drainage pressure of $p_{сл}$. The reduced pressure in this case will be raised to the $p_{сл}$:

$$p'_{ред} = \frac{P_{пр}}{f_п} + \frac{p_{сл} f_п}{f_п} = \frac{P_{пр}}{f_п} + p_{сл}.$$

After replacing in equation (59) of $p_{pc\partial}$ to $p'_{pc\partial}$, we will obtain

$$\Delta p_{a} = \frac{P_{np}}{f_{n}} + p_{cA} - p_{cA} = \frac{P_{np}}{f_{n}}.$$

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For a diagram in Fig. 105b, the reduced pressure and the pressure differential on throttle/choke 4 are determined taking into account the drainage pressure of the $p_{c\text{л}}$:

$$p_{ped} = \frac{P_{np}}{f_1 + f_6} + \frac{p_{c\text{л}} f_a}{f_a} = \frac{P_{np}}{f_1 + f_6} + p_{c\text{л}};$$

$$\Delta p_4 = \frac{P_{np}}{f_1 + f_6} + p_{c\text{л}} - p_{c\text{л}} = \frac{P_{np}}{f_1 + f_6},$$

where f_6 they is determined the sectional area of cylinder.

For an increase in the sensitivity of regulator, one should **waist** d_2 of plunger 2 (Fig. 105a) and increase diameter d_1 piston 1 whose relation frequently they lead to d_1/d_2 = 5 and more (in diagram in Fig. 105b this they reach by an increase in the diameter of piston 6). Furthermore, for an increase in the sensitivity one should the antifrictional of the moving elements of the regulator.

The values of p_{ped} under the mounting conditions of regulator

in the plum of the main line of hydraulic engine (Fig. 105) must be not more than 1-2 kgf/cm^2. For an increase in the stability of regulator against auto-oscillations in the way of the supply of liquid with the pressure of $p_{ред}$ to piston 1 (for a diagram in Fig. 105b - to piston 6) is established/installed damper throttle/choke) 5.

The antihunting circuit of the velocity of hydraulic motor by the setting up of throttle governor at output/yield won acceptance in machines with the oscillating (up to a sign change) load of hydraulic engine (in machine tools with the being changed cutting force in the period of the power stroke of hydraulic engine, etc.).

Figure 104b shows diagram of hydraulic engine control by means of changing the pressure of the liquid forced by the pump (see also Fig. 102d). The through section of the channel, which leads to the working chamber of the engine here is regulated by throttle/choke 2 of constant section. However, past this throttle/choke it passes into cylinder only the part of the liquid, which enters from pump, and remaining part is fused into the tank through the throttle/choke with self-adjustment. The abstract/removed part of the liquid is determined by the position of the floating gate of this throttle/choke 3, connected with small piston 1, to which from lower side is applied the force, proportional to the pressure of the liquid before throttle/choke 2 of constant section, and with upper is

applied the force, proportional to pressure after this throttle/choke (is proportional to operating pressure in hydraulic engine). In view of this depending on a change in the working load of hydraulic engine, the gate of automatic throttle governor, moving, changes the flow rate through it liquid into tank.

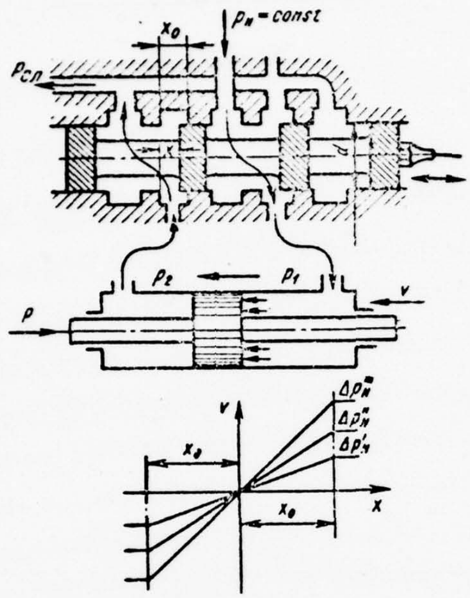
The pressure differential by throttle/choke 3, determined by the force of spring, usually does not exceed 2-3 kgf/cm^2.

For a preservation from excessively high braking effect, is applied ball bearing safety valve. For a decrease in the compressibility effect of the working medium, caused by the presence in it of the undissolved air, and provisions for a uniform outlet velocity from cylinder is established/installed support valve 4, adjusted to counterpressure 3-10 kgf/cm^2.

Throttle control by valve. In a series of systems and, in particular, servo systems, throttle/choke is the distribution valve, with the displacement of plunger of which they change the friction of expenditure (passage) slide valve ports.

Figure 106 gives diagram with the constant pressure of the power supply of the $(p_{н} = \text{const})$, in which as hydraulic engine is applied the actuating cylinder. In the position of the plunger of valve, shown on this diagram, the liquid through the right slide-valve window comes from pump into the right cavity of actuating cylinder, moving its piston to the left. Liquid from the left cavity of this cylinder is driven out through the left slide-valve window into drainage line.

Fig. 106. Diagram of throttle control by valve.



After designating a total pressure drop hearth by which works hydraulic system, through the $\Delta p_c = p_н - p_{сл}$, we can write, assuming that the friction of both windows are equal, and disregarding pressure in drainage line,

$$\Delta p_c = 2\Delta p_{ок} + \Delta p_н,$$

where the $\Delta p_н = p_1 - p_2 = \frac{P}{F}$ - the pressure differential, which corresponds to load P (friction we disregard): $\Delta p_{ок} = \frac{\Delta p_c - \Delta p_н}{2}$ - loss of pressure in one expenditure slide-valve window (slot); here F is an effective area of cylinder. With the adopted assumption of the equality of the friction of both expenditure windows, which corresponds to symmetrical valve, and neglecting the pressure of $p_{сл}$ in drainage line we have an $\Delta p_{ок} = p_н - p_1 = p_2$.

Fluid flow rate through the valve is equal not allowing for leakages and compressibility of liquid to the volume, described by piston:

$$Q = Fv, \text{ or } v = \frac{Q}{F},$$

where F and v is equal effective area and piston speed.

On the other hand, flow rate it is possible to express, according to expression (20), by

$$Q = \mu f \sqrt{\frac{2}{\rho}\Delta p_{o\kappa}} = \mu f \sqrt{\frac{2}{\rho}\cdot\frac{\Delta p_c - \Delta p_{\scriptscriptstyle H}}{2}} = \mu\pi x d \sqrt{\frac{1}{\rho}(\Delta p_c - \Delta p_{\scriptscriptstyle H})}, \qquad (60)$$

where f = πdx - the area of expenditure slot (window); x is discovery/opening slot.

In accordance with this, the piston speed

$$v = \frac{Q}{F} = \frac{\mu\pi d}{F}\sqrt{\frac{1}{\rho}(\Delta p_c - \Delta p_{\scriptscriptstyle H})}\, x = kx,$$

i.e. piston speed is the linear function of the displacement/movement of valve (or with the rectangular forms of slide valve ports - by the linear function of its discovery/opening). The curve/graph of dependence v = f (x) is called the discharge characteristic of valve. The standard characteristics of ideal valve with zero overlap (see Fig. 43c) with the different loads of hydraulic engine (with different $\Delta p_н$) are shown in Fig. 106.

Optimum relationship between the lost and total pressure. For the systems of throttle control, there is a optimum relationship between the lost and total pressure.

Let us determine the relationship between of $\Delta p_{ок}$ and $\Delta \bar{p}_н$, by which the power of hydraulic engine is maximum (friction of the line of gutter we disregard). With the flow rate of hydraulic system

Q the power of hydraulic engine

$$N = Q\,\Delta p_{\text{н}}\eta,$$

where the η - the efficiency of hydraulic engine.

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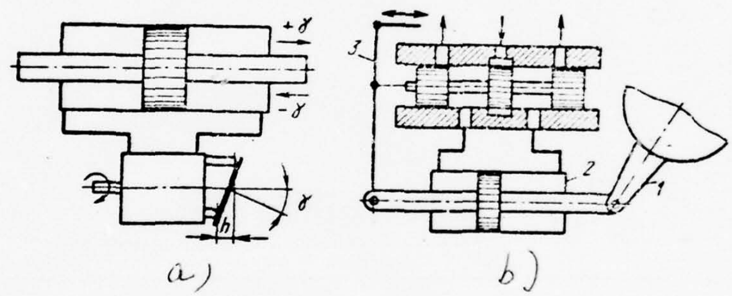

Fig. 107. Diagrams of the volumetric velocity control of hydraulic engine by a change in the feed of pump.

Flow rate, according to equation (60),

$$Q=c\sqrt{\Delta p_{ок}}=c\sqrt{\Delta p_c-\Delta p_н},$$

in accordance with which the last/latter equation

$$N=\eta C\sqrt{\Delta p_c-\Delta p_н}\,\Delta p_н.$$

Outer limit N

$$\frac{dN}{d(\Delta p_н)}=0=\sqrt{\Delta p_c-\Delta p_н}-\frac{\Delta p_н}{2\sqrt{\Delta p_c-\Delta p_н}},$$

whence

$$\Delta p_н=\frac{2}{3}\Delta p_c;\quad \frac{\Delta p_н}{\Delta p_c}=\frac{2}{3}.$$

From the last/latter expression, which characterizes the optimum relationship between of Δp_{H} and Δp_{c}, it follows that under the indicated conditions 1/3 pressures, developed with pump, it must be lost.

The efficiency of hydraulic system

$$\eta_c = \frac{Q}{Q_0}\frac{\Delta p_H}{\Delta p_c},$$

where Q_0 is a flow rate of pi idling.

Taking into account that

$$\frac{Q}{Q_0} = \sqrt{\frac{\Delta p_c - \Delta p_H}{\Delta p_c}},$$

we find the maximum efficiency

$$\eta_{c\,max} = \sqrt{\frac{\Delta p_c - \Delta p_H}{\Delta p_c}} \cdot \frac{\Delta p_H}{\Delta p_c} = \sqrt{1 - \frac{2}{3}} \cdot \frac{2}{3} \approx 0{,}39.$$

From the given analysis it follows that the systems of throttle control can be recommended to application/use only in the transmissions of small powers. In practice at powers more than 30 kW appear during the application/use of this control system of the difficulty of heat withdrawal.

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Velocity control of hydraulic engines by a change in the pump displacement.

The throttle control of the speed of hydraulic engine, used in systems with the pump of constant feed, is connected, as this was shown, with energy loss, and, if at the power-handling capacities of pump these losses were small, then with an increase in the powers

they reach inadmissible values.

In view of this at large powers (are more than 10-15 ~~h.s.~~ hp) and in the cases of the increased requirements for accuracy/precision and the stability of control, are applied the pumps with the adjustable feed (volumetric control).

In hydraulic drive with volumetric by control a change in the speed of hydraulic engine usually is realized by changing the pump displacement and thinner - the working volume of hydraulic motor. In comparison with the drive of throttle control, the drive with the adjustable displacment pump has higher efficiency of $\eta = 0.75-0.80$ (of the drive of the throttle control usually of $\eta = 0.3-0.4$), and also causes the less heating of liquid.

To the advantages of hydraulic drive with volumetric control, is related the fact that it has the rigid mechanical characteristic by which the speed of output shaft with a change in the load moment changes insignificantly.

Figure 107a depicts the schematic diagram of hydraulic drive with the rotor axial-reciprocating pump, the control of course of pistons in which is achieved by a manual or automatic change in the angle of the slope γ driving plate relative to the axle/axis of cylinder block. A change in the direction of the motion (reverse) of the piston of actuating cylinder here is achieved by a sign change of this angle.

The calculated feed of this pump

$$Q_н = qn = fznh = \frac{\pi d^2}{4} znh, \qquad (61)$$

where $q = fhz$ - pump displacement; here z is a number of suckers; $f = \pi d^2/4$ - piston clearance; n - rotation frequency in r/min; h — the adjustable pump lift (for the axial rotor- piston pump

$$h = D \operatorname{tg} \gamma);$$

in question here D - the diameter of a circle of the location of the axle/axes of cylinders in cylinder block; γ is angle of the slope of stator washer.

It is not difficult to see that the rated speed v of piston with an area of F in a similar diagram can smoothly vary within the limits of $\pm v_{max} = \pm \frac{Q_H}{F}$.

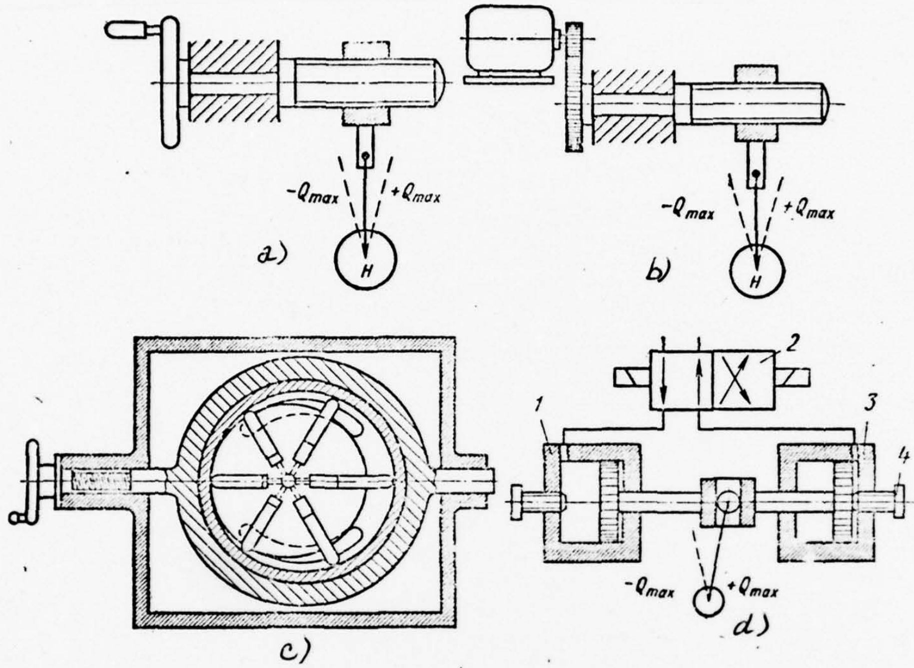
The characteristic of hydraulic drive of volumetric control in many respects depends on method and quality of the control mechanism of the feed of pump. This control, realized by hand or is automatically at the signals of different sensors, it is conducted by changing the pump displacement, which, in turn, usually is realized by means of the linear or angular mixing of the cell/elements with the aid of which changes the pump displacement.

Simplest it is the manual (Fig. 108a) either electromechanical (Fig. 108b) control, made with the aid of worm or helical vapor or another mechanisms.

Are common also the systems, into which the linear or angular displacement of the cell/element with the aid of which changes the

pump displacement, it is realized with the aid of hydraulic equipment/devices. If in the last/latter system it is required to ensure only the reverse of pump or motion in one direction at two speeds, and also motion in two directions at one speed of displacement in each direction, then are applied simple equipment/devices with two actuating cylinders 1 and 3 (Fig. 108d), which in the majority of cases are placed in pump casing. The control (limitation) of piston stroke is realized with the aid of mechanical (screw-type stop 4.

Fig. 108. Mechanisms of the volumetric feed control of pump.



For a control of cylinders, are commonly used distributors 2 with electromagnetic actuator.

When for the displacement of the adjustable cell/element are required large forces or it is required to ensure infinitely variable control, are applied servomechanisms (hydraulic boosters) with feedback.

The schematic diagram of the hydraulic assembly of the feed control of pump with a similar two-stage force is represented in Fig. 107b. Lever 1 with the aid of which is realized the regulating of pump displacement, is given with the aid of actuating cylinder 2 of slave/servo hydraulic booster. The inlet of 3 hydraulic boosters is connected with setting device and piston 2 with the lever of 1 feed control of pump.

Automatic feed control of pump. In practice are common the hydraulic systems with the pump of automatic feed control at the signals of different sensors (pressure, temperature, etc.).

In the hydraulic systems of machines, are common the control circuits on pressure (regulators of the ultimate pressure), by which the pressure, developed with pump, is utilized for the limitation of the feed of liquid or power, consumed by pump, up to the assigned minimum value.

The schematic diagram of axial-piston type automatically adjustable pump is shown in Fig. 109a. The spring of 2 regulators acts in the direction of the setting up of the inclined washer of 3 pumps at the position of the maximum angle γ the slope/inclination of this washer. The force of this spring resists pressure p of liquid at the output/yield of pump on the small piston of 1 regulator which upon achieving the assigned pressure compresses spring 2, decreasing this angle.

Consequently, regulator supports at the assigned pressure, determined by the precompression of spring 2, in practice constant supplies of liquid to the achievement of design pressure with the subsequent decrease in the feed, the intensity of growth in pressure depending on the characteristic of spring.

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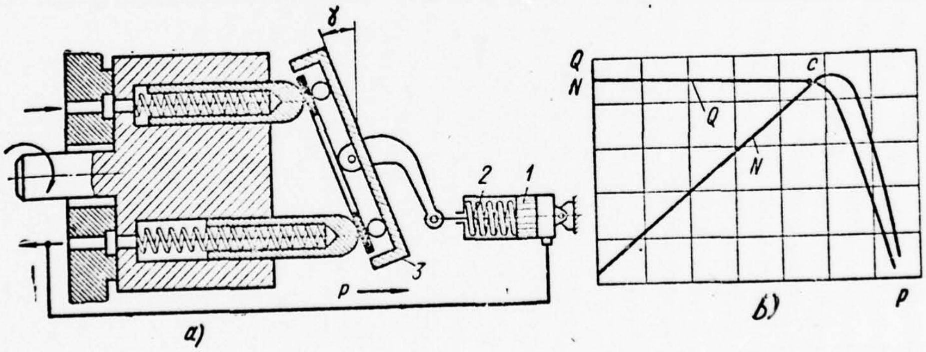

Fig. 109. Diagram of the automatic feed control of pump on pressure (a) and the dependence of feed Q and of power N of this pump on

pressure (b).

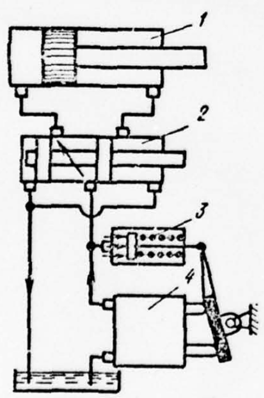

Fig. 110. Circuit diagram into the hydraulic system of pump with automatic feed control.

Figure 109b gives curves, that characterize feed Q and power N of pump in the function of pressure p at output/yield from pump. Point ~~from~~ C curved feed corresponds to the beginning of compression of spring of valve and to the beginning of the control (decrease) of the feed of pump. At certain feed pressure of pump will be lowered to zero. By the selection of the characteristic of the spring of regulator can be obtained the required for concrete/specific/actual target/purposes dependences $Q = f(p)$ and $N = f(p)$.

The circuit diagram of this pump 4 into hydraulic system with actuating cylinder 1 and distributor 2 is given in Fig. 110. During a pressure increase more than that established/installed the spring of regulator 3 is compressed, decreasing in this case the angle of the slope of the controlling washer of pump 4. The control of the piston of actuating cylinder 1 is realized in this diagram by distributor 2.

Pumps with a similar feed control (see Fig. 109), which obtained the name of the stabilizers of pressure, widely are applied also during maintenance by one pump of several periodically acting hydraulic engines (on hydroelectric station with the centralized power supply).

Systems with the reversive pumps of the adjustable feed. In this case, if by the construction of pump is provided for the possibility of the setting up of the controlling cell/element at the position, which corresponds to the negative (counting from the axis of the symmetry of cylinder block) value of controlled parameter (angle of the slope γ, etc.), a similar pump can ensure feed reverse in the same direction of rotation of its shaft. During the application/use of this pump, fails the necessity for distributor.

The schematic diagram of this hydraulic drive with actuating cylinder is shown in Fig. 111a. The reverse of hydraulic engine (in this diagram of actuating cylinder 1) and the control of its speed is realized by a change in direction and feed of pump 2. Since in this case are applied adjustable pumps 2 whose feed can smoothly vary within the limits of $\pm Q_{max}$, theoretically is represented possible to ensure reversing and the velocity control of hydraulic engine from zero to the maximum positive and negative value, determined by the volumetric parameters of pump and engine.

Fig. 111. Diagrams of volumetric control with reversive pumps.

Fig. 112. Diagrams of volumetric control with reversive pumps and feed pumps.

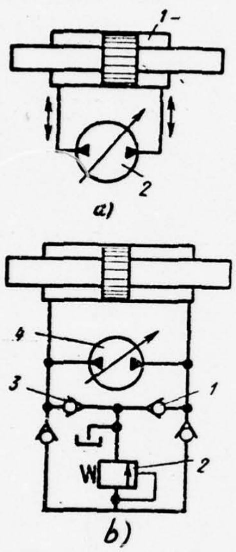

Fig. 111.

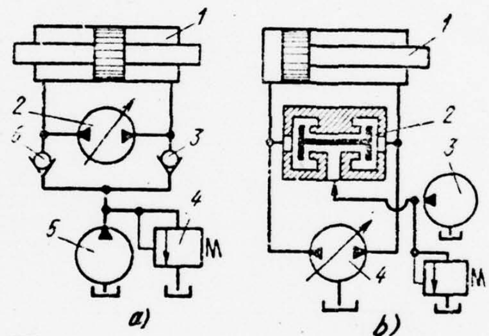

Fig. 112.

It is obvious, real hydraulic systems of this type they must be equipped by the supplementary equipment, which ensures the reliable power supply of pump by liquid and protecting devices (Fig. 111b). Specifically, for providing a makeup of the suction cavity of pump 4 system is supplied with check valves 1 and 3, which separate/liberate the working cavity of pump from that which suck in, and for a preservation from overload - by safety valve 2.

For an increase in the reliability of power supply system frequently they supply with supplementary feed pump 5 (Fig. 112a), that create certain, establish/installed by overflow valve 4, backwater in the line of the suction of basic pump 2. The working for this direction of the feed of pump cavity of cylinder 1 is intercept/detached from feed pump by 5 check valves 3 and 6.

During the application/use of cylinder 1 with one-sided stock/rod (Fig. 112b) diagram usually is equipped with dual return plate valve 2, which provides in work the flow of liquid, necessary for the compensation for a difference in the volumes of the cavities of cylinder 1. During the supply of liquid into any of the cavities of cylinder, the valve intercept/detaches this cavity from pump 3 makeups, connecting it through the forming slot with the opposite

cavity of cylinder. In this case, occurs either the makeup of pump by 4 booster pumps 3 (during the supply of liquid into the cavity of actuating cylinder, opposite to stock/rod), or the jettisoning of the excess of liquid into reservoir (during the supply of liquid into the rod cavity of actuating cylinder).

Control circuits with pump- dosing device. In the diagram of volumetric control examined above (see Fig. 107) the pump is simultaneously and power supply, and by speed governor. In this case it is establish/installed on the inlet into hydraulic engine.

However, in a similar setting up, in particular with low expenditures, are inherent the deficiency/lacks, noted in the examination of control circuits with setting up at the inlet of throttle/choke.

In view of this when it is necessary to ensure the bilateral hardness of hydraulic system, pump-regulator (flow-meter pump) establish/installs in the drain line of hydraulic engine. The power supply the systems by liquid provide with usually uncontrolled type supplementary special pump. The speed of hydraulic engine in this

case is determined by the volume of the liquid, take/selected from the last/latter pump by pump-regulator.

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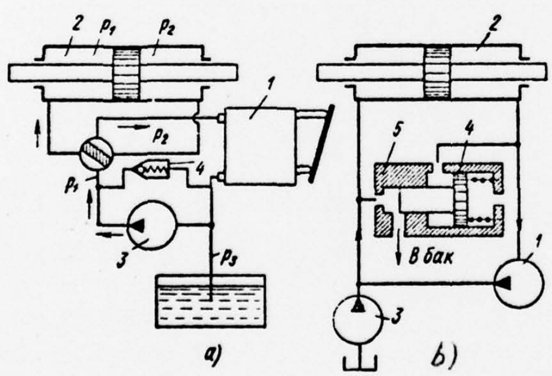

Fig. 113. Diagrams of volumetric control with pump-dosing device.
Key: (1). In tank.

The diagram of this system with the regulator, established/installed in the drain line of hydraulic engine, is given in Fig. 113a. The shaft of the adjustable flow-meter pump (regulator) 1 is strictly connected with the shafting of feeding (worker) pump 3 of the uncontrolled type whose feed they rely on fluid flow rate, somewhat exceeding consumption by its hydraulic engine 2 at its assigned speed. The surplus of liquid is recasted through overflow valve 4.

Because of a continuously variable change in the working volume of flow-meter pump 1, which works in this diagram in the mode/conditions of hydraulic motor, it is possible to ensure the infinitely variable control of amount of liquid, which enters the hydraulic engine (actuating cylinder) 2, whereupon the energy, which corresponds to the power of flow-meter pump 1, which works in this diagram as hydraulic motor, is not converted into heat, but returns to the shafting of pump 3.

The minimum value and the stability of the proportioned expenditure, and also the volumetric efficiency of pump depend on the load of engine, with change in which change the pressure and hydraulic slip.

From course "displacment pumps and hydraulic engines" it is known that the volumetric efficiency of pump

$$\eta_{o6} = 1 - \frac{\Delta Q_H}{Q_T},$$

where the Q_T and the ΔQ_H are a theoretical feel of pump and leakage in it.

Since the leakages in the pump of ΔQ_H in practice do not depend on the theoretical feed of Q_T, there is a limit of the minimum adjustable feed. With certain small working volume of the adjustable pump, 1 (it corresponds to the low value of Q_T) supply of liquid and, consequently, also the speed of hydraulic engine under load can as a result of hydraulic slipes completely cease itself. The latter will occur with of $\Delta Q_H = Q_T$.

From the diagram, given in Fig. 113a, it follows that with the no load of hydraulic engine 2 is correct condition $p_2 = p_1$, where p_1 is pressure of the adjustment of overflow valve 4 (friction also we disregard). In accordance with this, the pressure differential $\Delta p = p_2 - p_3$ in pump 1 with the no load of hydraulic engine 2 will be maximum and equal to $\Delta p = p_2 = p_1$. With the peak load of hydraulic engine 2, with which pressure p_2 in its nonoperative cavity is equal to zero, the pressure differential in pump 1 will be also equal to zero.

A deficiency/lack in the diagram, given in Fig. 113a, is the fact that the pressure of feeding pump 3 does not depend on the load of hydraulic engine; therefore at its small speeds (low pumping power) the considerable part of the power of this pump is converted into heat.

For a reduction in these losses, is applied the system whose diagram is given in Fig. 113b. The pump-regulator of 1 this diagram supplies liquid to the delivery line of feeding pump 3, equipped with the mechanism of an automatic change of the pressure depending on the load of hydraulic engine (actuating cylinder) 2. For this, is applied automatic throttle valve 5, established/installed on the forcing line

of feeding pump 3, which regulates the pressure of this pump depending on the counterpressure in the right (drainage) cavity of actuating cylinder 2, which is caused by its load.

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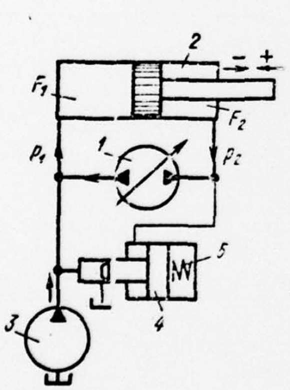

Fig. 114. Control circuits with differential braking valve.

The clear area of the stock/rod of the gate of braking valve 4 of this throttle/choke is selected the equal to half area of his piston stock/rod's on the part.

Operating principle of the system in question is based (not allowing for frictional forces and with the equality of the areas of the cavities of hydraulic engine) on the preservation/retention/maintaining of condition

$$p_1 + p_2 = \text{const.}$$

The force condition of equilibrium, which act in the engine:

$$p_1F = p_2F \pm P,$$

where p_1 and p_2 are pressure in worker and counterpressure and opposite (drainage) the cavities of actuating cylinder; P and F is load and the area of actuating cylinder.

If load P, applied to actuating cylinder (hydraulic engine 2), it acts against the direction of force of pressure p_1, then counterpressure p_2 will decrease, as a result the gate of braking valve 4 will be closed, increasing pressure at the output/yield of pump 3 to the value at which is provided the conlition $p_1 + p_2 =$

const. In the case when the load of actuating cylinder acts in opposite direction (in the line of force of pressure p_1), counterpressure p_2 increases, as a result braking valve is open/disclosed, decreasing pressure p_1.

Hydraulic system with the similar braking valve is applied in milling machines, to which are presented the requirements for the provision for an automatic velocity control of the motion (feed) of table depending on by a load change with cutting. The diagram of the hydraulic system of milling machine with a similar valve is represented in Fig. 114. Unlike the examined diagram, is here applied the actuating cylinder with one-sided stock/rod. System is equipped by two pumps 3 and 1, the first of which is the feeding pump and the second, that is the adjustable reversive pump, established/installed in the line of the gutter of actuating cylinder 2, it is speed governor. Pump 1 with the power stroke of cylinder 2 takes away liquid from the cavity of cylinder and supplies it to the delivery line of pump 3 whose pressure automatically is regulated with the aid of throttle valve 5 depending on the load of actuating cylinder 2 (by cutting forces). As in the valve, examined above (see Fig. 113b), the clear area of the small piston of valve 4 from the side of its shank usually is selected equal to the area of its shank.

The operating principle of the system in question entails the preservation/retention/maintaining of condition $p_1 + F_1/F_2\ p_2 =$ const.

The force condition of equilibrium, which act in the actuating cylinder:

$$p_1F_1 = p_2F_2 \pm P,$$

where p_1 and p_2 are pressure in worker and counterpressure in opposite the cavities of actuating cylinder; F_1 and F_2 is an area of the cavity of actuating cylinder; P of the load of actuating cylinder.

In such a case, when load P acts against the direction of force of pressure p_1, counterpressure p_2 will decrease. As a result the gate of valve 4 will be closed, increasing pressure p_1 to the value by which will be restore/reduced the condition

$$p_1 + \frac{F_1}{F_2} p_2 = \text{const.}$$

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